Vial

Irregularities in Braking Power of Freight Cars with Special Reference to Wheel Service

IRREGULARITES IN BRAKING POWER OF FREIGHT CARS WITH SPECIAL REFERENCE TO WHEEL SERVICE

BY

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THESIS

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I HEREBY RECOMMEND THAT THE THESIS PREPARED BY

F. K. VIAL

ENTITLED IRREGULARITIES IN BRAKING POWER OF FREIGHT CARS WITH SPECIAL REFERENCE TO WHEEL SERVICE

BE ACCEPTED AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE PROFESSIONAL DEGREE OF Civil Engineer

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IRREGULARITIES IN BRAKING POWER OF FREIGHT CARS

WITH SPECIAL REFERENCE TO WHEEL SERVICE

The vast amount of thoughtful consideration that has been given to the subject of automatic air brakes for railway equipment has brought this mechanism practically to perfection. There are however, many items between the triple valve and the brake shoe that have escaped critical analysis, and the improper adjustment that is frequent in these parts leads to a serious interference in the final result that is so much desired, namely, uniform distribution of brake effort proportional to the requirement of each car of the train.

The triple valve may be perfect and the braking ratios to the wheels calculated to a nicety, and everything in perfect working condition, yet the result may be upset by so small an item as the position of the brake hanger, piston travel, improper brake lever applied when repairs are made and wide variation in percentage of braking power based on gross load. Indeed, some of these items pass entirely unnoticed and not only throw the whole brake mechanism out of balance, but are expensive in the way of broken truck pedestals, deformed hangers, deformed brake beams, excess brake shoe consumption, brake burned and slid flat wheels, bent levers, break-in-twos, buckled cars, etc.

In order to discover and clearly point out the evil effect of irregularities in individual cars and in present standards with reference to braking power, it is necessary to review in detail the fundamental principles of retardation, the application of which will not only definitely locate undesirable and expensive errors but will with equal clearness point to the remedy. Therefore, the principles of braking will be discussed in detail and full data given to easily handle any problem in acceleration or retardation.

The braking power of a car is the ratio of the total shoe pressures to the tare weight, for example:

The tare weight of a car is 40,000 lbs. and when brakes are fully applied has a shoe pressure of 3,000 lbs. on each of the eight wheels, making a total of 24,000 lbs. for the car. What is its percentage of braking power?

$$\text{Percentage of braking power} = \frac{\text{total shoe pressure}}{\text{weight of car}} = \frac{24000 \text{ lbs}}{40000 \text{ lbs}} = 60\%$$

$$\text{Braking power per wheel is } \frac{24000 \text{ lbs}}{8} = 3000 \text{ lbs.}$$

The ordinary air pressure carried in the train line is 70 lbs. and the auxiliary reservoirs have been made so that the air which they contain at 70 lbs. pressure will equalize at 50 lbs. pressure when connection is made with the brake cylinder and the piston travels eight inches.

It has been found by experience that the best results are obtained when the braking power on the wheel does not greatly exceed 60% of its minimum load. Therefore, the M. C. B. Association has adopted the rule that all freight equipment should be braked at 60% of its tare weight when the brake cylinder pressure is 50 lbs. This standard

fixes the relation between the volume of the auxiliary reservoir and the brake cylinder, because the initial pressure in the auxiliary reservoir shown in Fig. 1 is 70 lbs. and is controlled by the triple valve. When a service application is required, the triple valve opens the passage way between the auxiliary reservoir and brake cylinder, allowing the air to enter the brake cylinder and press the piston out eight inches at which point the brakes are fully set and the piston can travel no farther. The air that was contained in the auxiliary reservoir now occupies an enlarged space, represented by the volume of auxiliary reservoir, the air cylinder and connecting passage ways. When the air is equalized throughout this enlarged space, the relation should be such that the air pressure is 50 lbs.

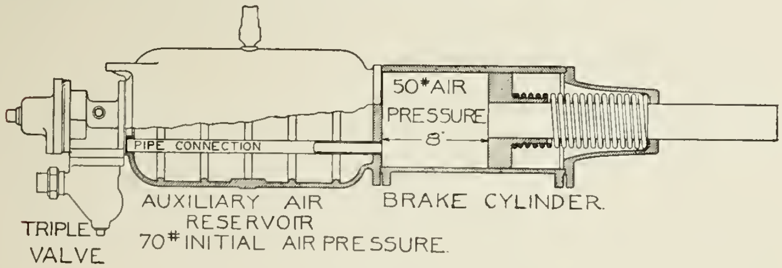


Fig. No. 1.

The pressure per sq. in. of the air is inversely proportional to the size of the receptacle in which it is placed, but before this proportion can be applied, however, it is necessary to note the difference between atmospheric pressure and absolute pressure. Free air of the atmosphere is under a constant pressure of approximately 15 lbs. per sq. in., which cannot be shown on a pressure gauge. The absolute pressure is made up of the ordinary gauge pressure plus 15 lbs. atmospheric pressure. Thus, the absolute pressure in the auxiliary reservoir becomes 85 lbs. and the absolute pressure of equalization is 65 lbs.

The relation between the volume of the auxiliary reservoir and the brake cylinder at 8 inches piston travel is fixed by the drop in pressure of 20 lbs., the volume occupied by the air being inversely proportional to the pressure. Thus, if we assume the auxiliary reservoir to equal 100%, the combined volume of the auxiliary reservoir and brake cylinder will be 85/65 of 100 or 130.77%. In other words, the volume of the brake cylinder is represented by the 30.77% increase, which, divided into 100% shows that the auxiliary reservoir has 3.25 times the volume of the brake cylinder. In order to allow for leakage through triple valve, clearance and other losses, it is customary to make the auxiliary reservoir from three and one-quarter to four and one-half times the volume of the brake cylinder. In case 90 lbs. of air is carried in the train line, the pressure in the brake cylinder at the point of equalization for 8 inches travel would be

$$\frac{90 \text{ plus } 15}{130.77} - 15 = 65.4.$$

This indicates that the braking power can be increased 30% for full service applications if the pressure in the auxiliary cylinder is raised to 90 lbs.

The old rule for braking power was 70% of the tare weight of the car at 60 lbs. cylinder pressure. Under ordinary working conditions, this air pressure in brake cylinder is never obtained so the rule was changed to 60% at 50 lbs. pressure. This change raises the brak-

ing power 1 lb. for every 35 lbs. or practically 3%. This is shown by the application to a car weighing 30,000 lbs.

Old Rule.

Braking power—70% of 30,000 lbs. = 21,000 lbs.
Pressure delivered by brake cylinder—50.3 x 60 = 3,000 lbs.
Brake leverage ratio $\frac{21,000}{3,000} = 7$

New Rule.

Braking power—60% of 30,000 lbs. = 18,000 lbs.
Pressure delivered by brake cylinder—50.3 x 50 = 2,500 lbs.
Brake leverage ratio $\frac{18000}{2500} = 7.2$

This shows that the brake leverage is increased from seven times the brake cylinder pressure to 7.2 times, and if the air pressure is raised to 60 lbs. in the brake cylinder the new shoe pressure becomes 3000 x 7.2 = 21600 instead of 21000 lbs. as originally, representing an approximate increase of 3%.

Diameter of cylinder	8″	10″	12″	14″	16″	18″
Area of Piston	50.3	78.5	113.1	153.9	201.1	254.4
Tot. Pressure on Piston at 50 lbs. Cyl. Press.	2500	3900	5650	7700	10050	12700
Vol. at 8″ P. T. cu. in.	402.4	628.0	904.8	1231.2	1608.8	2036.0
Vol. of aux. tube and Cylinder Clearance	48	48	48	48	48	48
Vol. of Aux. Res. Cu. In.						
Truck	1180	1774	2457	....	....	....
Tender	1476	2457	3096	4476	5724	....
Passenger	1476	2457	3096	4476	5724	7436
Driver	2145	3096	4476	5724	7436	8577
Freight	1620	2800	....	....	....	....
Ratio-Aux. Res. to brake Cylinder 8″ Travel.						
Truck	2.62	2.62	2.58	....	....	....
Tender	3.28	3.64	3.25	3.50	3.46	....
Passenger	3.28	3.64	3.25	3.50	3.46	3.57
Driver	4.77	4.58	4.70	4.47	4.49	4.12
Freight	3.60	4.15	....	....	....	....

TABLE No. 1.

The relations between brake cylinder, auxiliary air cylinder, piston travel and air pressures for various classes of equipment are shown in Table No. 1

	8″ Brake Cylinder Freight Equipment, Train Pipe Reductions.										10″ Brake Cylinder FreightEquipment, Train Pipe Reductions.									
	5 lbs.		10 lbs.		15 lbs.		20 lbs.		25 lbs.		5 lbs.		10 lbs.		15 lbs.		20 lbs.		25 bs.	
	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90	70	90
2″ P.T.	40	40	63	82	63	82	63	82	63	82	46	46	64	83	64	83	64	83	64	83
4″ ʻʻ	19	19	52	52	59	77	59	77	59	77	17	17	55	55	60	78	60	78	60	78
6″ ʻʻ	9	9	33	33	55	56	55	72	55	72	5	5	32	32	57	59	57	74	57	74
8″ ʻʻ	4	4	22	22	40	40	51	58	51	67	..	..	19	19	40	40	53	61.	53	70
10″ ʻʻ	1	1	16	16	30	30	45	45	48	59	..	..	12	12	28	28	45	45	50	62
12″ ʻʻ	..	..	11	11	23	23	36	36	46	48	..	..	6	6	20	20	35	35	48	49

NOTE:—**Bold Face** figures indicate equalization pressures.

TABLE No. 2.

Approximate pressures in brake cylinder with various train pipe reductions and piston travel, from 70 lbs. to 90 lbs. initial auxiliary pressures.

FOUNDATION BRAKE GEARS.

The foundation brake gears are designed with the proper ratio at the time the car is built, to produce 60% braking power when the cylinder pressure is 50 lbs., and on account of the dissimilarity in tare weights, there can be no fixed standard of levers for each capacity of car. The correctness of the braking leverage is seldom questioned after cars have been placed in service, although it is well known that there are great discrepancies in individual cars, the braking powers varying from 50% to 90%. It is therefore essential that more attention should be given to checking braking power of cars and inspectors should be posted on easy methods of making the calculations. At first sight this problem may look rather difficult, but a little study will simplify the whole question, and as this is a subject which should be well understood by everyone having the care of cars under his jurisdiction, it will be discussed in detail.

By means of the brake rigging, the primary force in the brake cylinder is increased, divided into eight parts and applied to the wheels in the form of shoe pressure. The mechanical advantage of the brake rigging is based on the law of levers. Therefore, to fully understand the method of calculating brake pressures, the law of levers should be thoroughly mastered. There are five items that require attention in considering levers:

One: Power
Two: Load or Resistance
Three: Fulcrum
Four: Power Arm
Five: Load Arm

There are three classes of levers, depending upon the position of the fulcrum:

Class I Fulcrum between power and load.
Class II Load between power and fulcrum.
Class III Power between load and fulcrum.

In each class, the tendency of the lever is to turn on the fulcrum, and the intensity of this tendency is represented by the amount of force multiplied by its distance from the fulcrum. This product in mechanics is called the turning moment. When applied to the power it is called the power turning moment, and when applied to the load it is called the load turning moment. If the lever is to remain in equilibrium, the turning moments must be equal and opposed to each other. In other words, the input and output must balance; the power multiplied by the power arm being the input; and the load multiplied by the load arm being the output.

CLASS I.

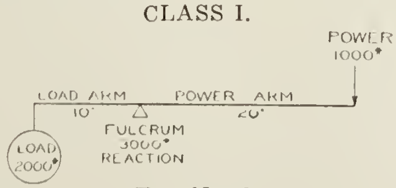


Fig. No. 2.

Figure No. 2 illustrates a lever of the first class because the fulcrum lies between the power and the load.

To illustrate solving the various forces in a lever of this class, assume the power or force of 1000 lbs. to be 20 in. from the fulcrum as shown in the figure, the load or resistance being on the opposite side and 10 in. from the fulcrum. Find the load that would hold the power in equilibrium.

According to the laws of levers, the power turning moment equals the load turning moment. Then in this case:

Power turning moment = 1000 lbs. x 20 in. = 20000 in. lbs.
Therefore, 20000 in. lbs. = Load Turning Moment
And the load $= \frac{20000}{10} =$ 2000 lbs.

In the above example, the 20000 in. lbs. is balanced by the turning moment at the other end of the lever, in which the lever arm is 10 inches long. Hence each pound at the load end will have a turning moment of 10 in. lbs., and it will require as many pounds load to balance the 1000 lbs. power as 10 is contained in 20000 or 2000 lbs. This means that the power of 1000 lbs., 20 inches from the fulcrum will just balance 2000 lbs. load 10 inches from the fulcrum.

The load on the fulcrum is equal to the sum of the power and load because both are acting downward with the fulcrum between them. In this case the downward force on the fulcrum is 3000 lbs.

In case of movement, the relative distance traveled by the power and load is proportional to the length of their respective lever arms. In the above case, the power would move twice as far as the load because its lever arm is 20 inches, while that of the load is 10 inches. The amount of work performed is the power multiplied by the distance through which it travels, thus, if the power should move the lever downward 6 inches, the work done would be 1000x6 inches equals 6000 inch lbs. The load would move 3 inches and the resistance overcome would be 2000x3 inches or 6000 inch lbs., which represents the work accomplished.

CLASS II.

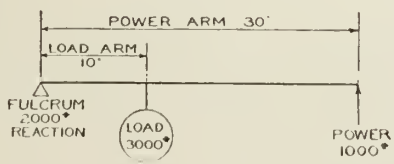


Fig. No. 3.

Figure 3 illustrates a lever of the second class because the load is between the power and the fulcrum.

Adhering to the law defined in Class I, the analysis of the forces is as follows:

Power turning moment = 1000 lbs. x 30 in. or 30000 in. lbs.
Therefore, 30000 in. lbs. = Load Turning Moment
And the Load $= \frac{30000}{10} =$ 3000 lbs.

In the above example each pound of load has a turning moment about the fulcrum of 10 in. lbs. Hence, the load equals $\frac{\text{Power Turning Moment}}{10} =$ 3000 lbs. Or, arrived at differently—each pound of power has a turning moment about the fulcrum of 30 in. lbs.

As this is three times the turning moment of each pound of load, the power can be but one-third of the load.

In this case we have the fulcrum at the end of the lever. The force acting on the fulcrum is then the difference between the load, which exerts a force in one direction, and the power, which exerts a force in the opposite direction, or 2000 lbs. In case the power should move 6", the work done would be 1000 x 6 inches or 6000 inch lbs. The load lever arm being but one-third as long as the power arm, would have a movement of 2", so that the resistance overcome would be 3000x2" equals 6000 in. lbs.

CLASS III.

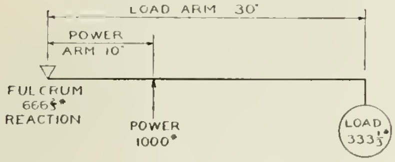


FIG. No. 4.

Figure No. 4 illustrates a lever of the third class because the power is between the load and the fulcrum.

In this case the equation becomes:—

Power turning moment = 1000 lbs. x 10 in. or 10000 in. lbs.

Therefore, 10000 in. lbs. = Load Turning Moment

And the Load $= \frac{10000}{30}$ or 333⅓ lbs.

Each pound of load has a turning moment about the fulcrum of 30 in. lbs., as compared with a turning moment of 10 in. lbs. for each pound of power. Hence the load must be one-third of the power.

As in Class II, we here have the fulcrum at the end of the lever. The force exerted on the fulcrum is therefore the difference between the power, which exerts a force in one direction, and the load, which exerts an opposing force, or 666⅔ lbs. In this case, if the power should move 2", the work done would be 1000x2" equals 2000 in. lbs. The load lever arm being three times greater than the power arm the load would move 6" and the resistance overcome would be 333⅓x6" equals 2000 in. lbs.

In the foregoing analyses of the various classes of levers the following laws have been established:

The power multiplied by the power arm (called the power turning moment) equals the load multiplied by the load arm (called the load turning moment.) Or, stated in another way,—the power and load are to each other inversely as the length of their respective lever arms. In applying these laws to the calculation of foundation brake rigging, the first point to be considered is the pin on which the lever has a tendency to turn when the power is applied. This is the fulcrum and the point from which measurements are taken to establish the lever arms. The force applied corresponds to the power and the resistance corresponds to the load. Multiplying the known force by its distance from the fulcrum and dividing this product by the distance from the fulcrum to the point of resistance gives as a quotient the maximum resistance the force can overcome. With this principle thoroughly understood, it becomes an easy matter to trace any force from pin to pin

and note the final pressure at any point, and calculate the travel of any rod or pin.

From the above discussion, it is seen that a small force with a long lever arm can overcome a large force with a small lever arm. The amount of work, however, is always exactly alike at each end of the lever. In mechanics, work is expressed as a force exerted over a given distance. Work, then, is composed of the elements force exerted and distance through which motion occurs, neither of which in themselves constitute work.

Any force, no matter how large, exerted against an object that does not move, does no work, and, conversely, any force moving thru space without resistance is doing no work. In each case, however, the force exerted against the immovable body or the body moving through space without resistance, represents a potential work which might result in useful work if differently applied.

It will be noted in each case of a lever that the power multiplied by the distance which it moves through, is exactly equivalent to the load multiplied by the distance which it moves. This means that the amount of work performed by a given force cannot be increased by any system of levers for the simple reason that when the load moved is large, the distance through which it is moved is correspondingly shortened, and the product of the load and the distance through which it moves is always constant, regardless of the system of levers or any other form of mechanical advantage that can be employed.

In any system of brake levers, the magnitude of the original force is the area of the piston in the brake cylinder multiplied by the pressure per sq. in. which, for an 8" cylinder the maximum is 50x50 lbs. or 2500 lbs. This force multiplied by 8", the normal piston travel, gives a quantity of 20000 in. lbs. of potential work, one-half of which is applicable to each end of the car. This quantity is absolutely constant for if we determine the force at any pin and multiply it by the distance through which it travels, the product will be 20000 in. lbs. divided by the portion of the car actuated. The cylinder lever which actuates the whole car must produce 20000 in. lbs. of work. The intermediate levers actuating one-half the car must develop 10000 in. lbs. of work, and the fixed truck lever actuating but one brake beam must perform 5000 in. lbs.

This law determines at once the distance that the shoe moves for any ratio of brake leverage. For example,—if the braking power of a car having a cylinder pressure of 2500 lbs. and an 8" piston travel is 25000 lbs., the distance traveled by the brake shoe will be:

$$\frac{20000 \text{ in. lbs. potential work}}{25000 \text{ lbs. braking power}} = .8 \text{ inch}$$

Or cylinder pressure of 2500 lbs. is one-tenth of the braking power of 25000 lbs., therefore the distance traveled by the brake shoe must be one-tenth of 8 inches or .8 inch.

Multiplication of the power of the brake cylinder by 10 through the foundation brake is not good practice on account of the very small shoe movement, requiring accurate and continuous adjustment to get the proper piston travel. The M. C. B. rules now call for the maximum multiplication by leverage of 9, but a ratio of 7 or 8 to 1 is still better, allowing 1" or a little more for shoe movement.

Several different types of foundation brakes are shown in Figures Nos. 5 and 6, showing the pull on each connecting rod and the relation between the cylinder pressure and the pressure in each part of the brake system until the shoe pressure is finally developed.

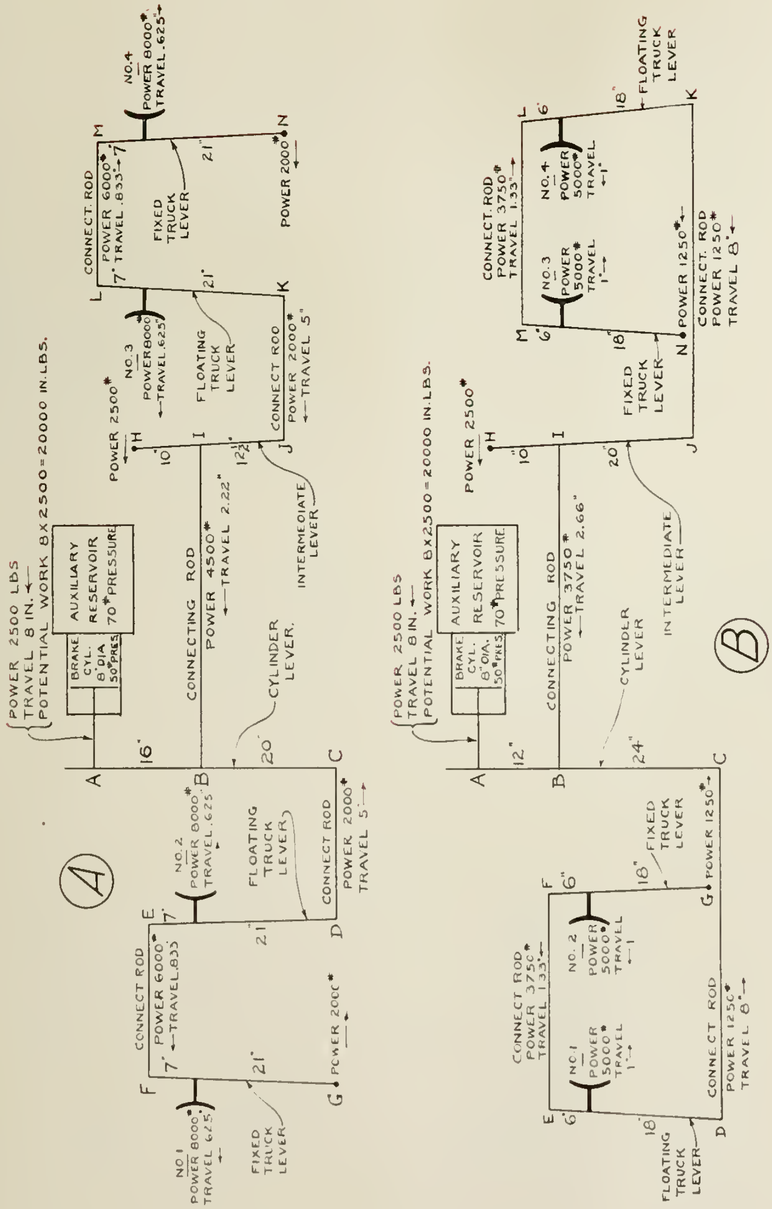
Figure 7 shows in detail the M. C. B. standards for brake levers. These are shown merely as a matter of reference.

The M. C. B. Assn. allows the following stresses to be used in foundation brake gears in passenger service, which are also applicable to freight service:

Levers	23000 lbs. per sq. in.
Rods	15000 lbs. per sq. in.
Jaws	10000 lbs. per sq. in.
Pins	10000 lbs. per sq. in.

Pin bearing, projected area, 23000 lbs. per sq. in.

The stresses are to be calculated for the full 70 lbs. cylinder pressure. The M. C. B. recommend that the 8" cylinder be limited to cars weighing 37000 lbs. and under; that the 10" cylinder be used on cars weighing between 37000 lbs. and 58000 lbs.



A
POWER 2500 LBS
TRAVEL 8 IN.
POTENTIAL WORK 8X2500=20000 IN. LBS.
BRAKE CYL. 8" DIA 50# PRES.
AUXILIARY RESERVOIR 70# PRESSURE
CONNECTING ROD
POWER 4500#
TRAVEL 2.22"
CYLINDER LEVER.
INTERMEDIATE LEVER
POWER 2500#
NO. 3
POWER 8000#
TRAVEL .625"
FLOATING TRUCK LEVER
CONNECT ROD
POWER 2000#
TRAVEL 5"
CONNECT. ROD
POWER 6000#
TRAVEL .833"
FIXED TRUCK LEVER
POWER 2000#
NO. 4
POWER 8000#
TRAVEL .625"
NO. 2
POWER 8000#
TRAVEL .625"
NO. 1
POWER 8000#
TRAVEL .625"
B
POWER 3750#
TRAVEL 2.66"
CONNECT. ROD
POWER 3750#
TRAVEL 1.33"
POWER 5000#
TRAVEL 1"
POWER 1250#
TRAVEL 8"

Fig. No. 5.

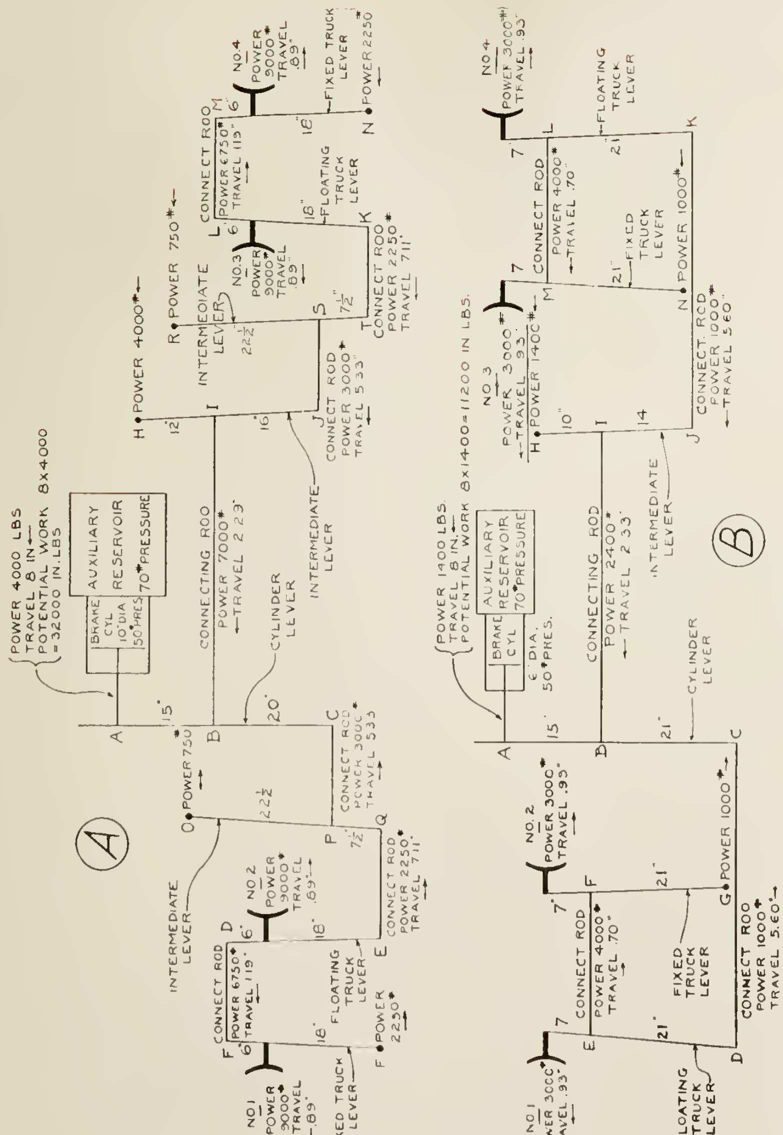
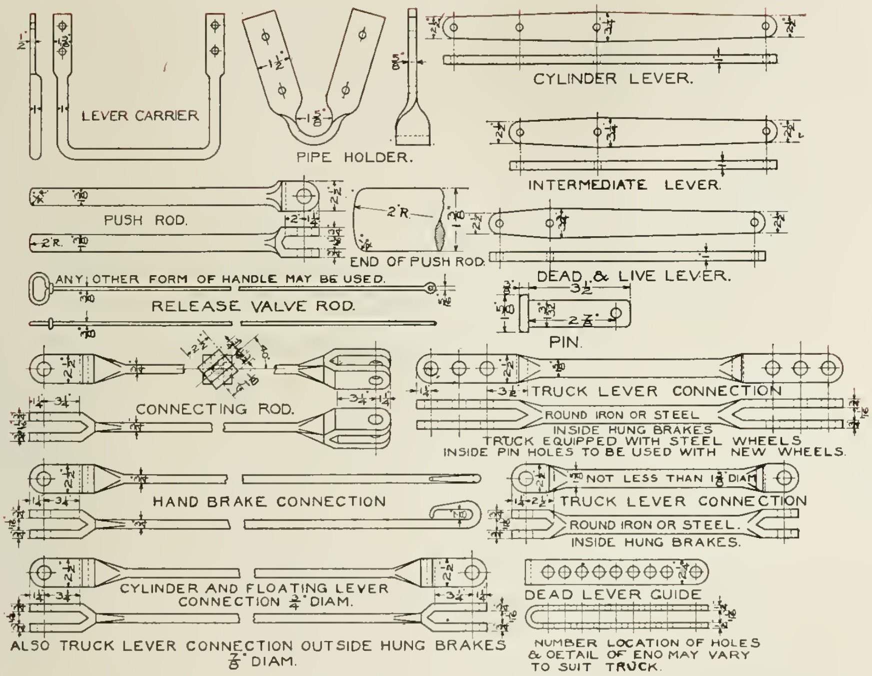

A
POWER 4000 LBS
TRAVEL 8 IN
POTENTIAL WORK 8x4000
=32000 IN. LBS
BRAKE CYL 10" DIA 50# PRES.
AUXILIARY RESERVOIR 70# PRESSURE
CONNECTING ROD
POWER 7000#
TRAVEL 2.23"
CYLINDER LEVER
INTERMEDIATE LEVER
CONNECT ROD
POWER 3000#
TRAVEL 5.33"
CONNECT ROD
POWER 2250#
TRAVEL 7.11"
FLOATING TRUCK LEVER
FIXED TRUCK LEVER
NO 1
POWER 9000#
TRAVEL .89"
NO 2
POWER 9000#
TRAVEL .89"
NO 3
POWER 9000#
TRAVEL .89"
NO 4
POWER 9000#
TRAVEL .89"
CONNECT ROD
POWER 6750#
TRAVEL 1.19"
POWER 750
POWER 4000#
POWER 2250
B
POWER 1400 LBS.
TRAVEL 8 IN.
POTENTIAL WORK 8x1400=11200 IN LBS.
BRAKE CYL 6" DIA. 50# PRES.
AUXILIARY RESERVOIR 70# PRESSURE
CONNECTING ROD
POWER 2400#
TRAVEL 2.33"
CYLINDER LEVER
INTERMEDIATE LEVER
CONNECT ROD
POWER 4000#
TRAVEL .70"
CONNECT ROD
POWER 1000#
TRAVEL 5.60"
FIXED TRUCK LEVER
FLOATING TRUCK LEVER
NO 1
POWER 3000#
TRAVEL .93"
NO 2
POWER 3000#
TRAVEL .93"
NO 3
POWER 3000#
TRAVEL .93"
NO 4
POWER 3000#
TRAVEL .93"
POWER 1000#
POWER 1400#

FIG. No. 6.



NOTE:—For cars weighing between 37,000 lbs. and 58,000 lbs. 10″ brake cylinders are used, and for cars weighing less than 37,000 lbs. 8″ cylinders are used.

Fig. No. 7.

In calculating the forces in the various parts of the brake rigging that are transmitted from the brake cylinder piston to the brake shoe under the conditions shown in Fig. 5, the first step is to note the force exerted by the brake cylinder piston for a full service application when the piston travel is 8″. We find this quantity to be 2500 lbs.

Taking up each lever shown in Fig. 5-A separately, we have the following:

Cylinder Lever A-C.

1st: To ascertain the force which the cylinder lever delivers to connecting rod C-D at C. This lever is of Class I, explained on Pages 6-7, with the fulcrum at the point B, hence:

Power turning moment = 2500 x 16 = 40000 in. lbs.
Load lever arm is 20 inches

$$\text{Pull at pin C} = \frac{40000}{20 \text{ in.}} = 2000 \text{ lbs.}$$

This indicates a 2000 lb. pull in connecting rod C-D.

2nd: To find the force in connecting rod B-I. In this case the fulcrum is at C and the lever becomes a Class II lever, therefor:

Power turning moment = 2500 x 36 = 90000 in. lbs.
Connecting rod lever arm is 20 inches.
Pull at pin B $= \dfrac{90000 \text{ in. lbs.}}{20 \text{ in.}} = 4500$ lbs.

Floating Truck Lever D-E.

The pressure delivered at brake beam is delivered by the brake lever D-E, having its fulcrum at E, and a 2000 lb. pull at D. This represents a lever of Class II, hence:

Power turning moment = 2000 lbs. x 28 in. = 56000 in. lbs.
Brake beam lever arm is 7 inches
Pressure at Beam No. 2 $= \dfrac{56000 \text{ in. lbs.}}{7 \text{ in.}} = 8000$ lbs.

Pressure at each brake shoe $= \dfrac{8000 \text{ lbs.}}{2} = 4000$ lbs.

To determine the pressure delivered to the connecting rod E-F, the floating lever D-E having its fulcrum at Beam No. 2 becomes a lever of Class I, hence:

Power turning moment = 2000 lbs. x 21 in. = 42000 in. lbs.
Connecting rod lever arm is 7 inches
Pressure at pin E $= \dfrac{42000 \text{ in. lbs.}}{7 \text{ in.}} = 6000$ lbs.

The pressure of 6000 lbs. is delivered to connecting rod E-F which actuates Beam No. 1.

Fixed Truck Lever F-G.

The pressure in Beam No. 1 is delivered by the truck lever F-G, having its fulcrum at G and a pressure of 6000 lbs. at F, being a Class II lever, hence:

Power turning moment = 6000 lbs. x 28 in. = 168000 in. lbs.
Brake beam lever arm is 21 inches
Pressure at Beam No. 1 $= \dfrac{168000 \text{ in. lbs.}}{21 \text{ in.}} = 8000$ lbs.

Pressure at each brake shoe $= \dfrac{8000 \text{ lbs.}}{2} = 4000$ lbs.

Having calculated the shoe pressures at the cylinder end of the car it is necessary to check the intermediate lever H-J to prove that the shoe pressures at the other end of the car correspond with those just calculated.

Intermediate Lever H-J.

In this case the fulcrum is at H with a 4500 lb. pull at I which are the conditions of a lever of Class III, hence:

Power turning moment = 4500 lbs. x 10 in. = 45000 in. lbs.
Connecting rod lever arm = 22½ inches
Pull at pin J $= \dfrac{45000 \text{ in. lbs.}}{22\frac{1}{2} \text{ in.}} = 2000$ lbs.

Since this is the same pull as was found for connecting rod C-D and the truck levers are alike at both ends of the car we know that the shoe pressures are the same.

The total braking pressure of the car is

4000 lbs. x 8 = 32000 lbs.
Weight of car = 50000 lbs.
Percentage of braking power = $\frac{32000}{50000} = 64\%$

For the purpose of checking, several observations can be made.

Assume the cylinder lever A-C to be suspended at B by the connecting rod B-I, having 2500 lbs. suspended at A and 2000 lbs. at C. It is self-evident that the connecting rod must carry the sum of these two items or 4500 lbs. Similarly, if the floating truck lever D-E is balanced on brake Beam No. 2 supporting 6000 lbs at E and 2000 lbs. at D, the pressure on the beam will be the sum of these two items or 8000 lbs. Observations such as these quickly detect any errors in calculation.

On Page 6 the law was established that a system of leverage can increase the initial force, which in this case represents the push of the piston of the brake cylinder by any desired amount at the expense, however, of distance moved. This law is checked in the case of brake rigging as shown in Fig. 5-A as follows:

Initial force = 2500 lbs.
Distance traveled = 8 inches.
Potential work = 2500 lbs. x 8 inches = 20000 in. lbs.

This is divided into four parts so that the potential work at each brake beam is 5000 in. lbs. and since the connecting rods C D and J-K each actuate one end of the car, the potential work would be 10000 in. lbs. The check is as follows:

In the cylinder lever A-C, the movement at A is 8 inches. The movement at C if B were a fixed point would be inversely as the lever arms or $\frac{8 \times 20}{16} = 10$ inches, but the connecting rod B-I is floating so that one-half of the movement is transferred to the intermediate lever H-J. Therefore, the momevent at C is one-half of 10 inches or 5 inches.

Pull on connecting rod C-D = 2000 lbs.
Travel = 5 in.
Potential work = 2000 lbs. x 5 in. = 10000 in. lbs.

Since 10000 in. lbs. is one-half of the potential work of the brake cylinder, the calculations involved are proven correct.

Examining the floating truck lever D-E by the same method, we have the following:

Movement at D = 5 inches.
Movement at E if brake beam No. 2 were fixed would be inversely as the lever arms 21 and 7 or
$\frac{5 \times 7}{21} = 1.66$ plus inches.

But, the lever D-E being floating, one-half of this movement is required to press the brake beam No. 1 against the wheel. Therefore, the actual movement which remains at E to press the beam against the wheel is one-half of 1.66 inches or .83 inch.

The connecting rod E-F therefore has a travel of .83 inches, having a pressure of 6000 lbs. This represents the potential work of 6000 x.83 or 5000 lbs. This proves the calculation so far to be correct, because there are four brake beams and 5000 lbs. is one-quarter of the potential work at the brake cylinder which is required to actuate one brake beam.

In the fixed truck lever F-G, the movement at F is .83 inch, the lever being fixed at G, the movement at brake beam No. 1 is inversely as the lever arms 28 and 21, or $\frac{.83 \times 21}{28} = .625$ inch travel of brake beam No. 1

The pressure on this brake beam being 8000 lbs. and the travel .625 inch, the potential work becomes 5000 lbs. for one quarter of the car, which proves the calculations throughout to be correct.

Having shown the transmission of power from brake cylinder to brake shoe in detail it can readily be seen that a short cut can be used in every case without going through the intermediate operations. It is only necessary to examine each lever in the system and note which end receives the power and which end delivers the force, then, regardless of the number of levers, the original power multiplied by all the receiving arms must equal the final pressure multiplied by all the delivering arms. Or in other words, the input and output must be equal. This rule applied to Figures 5 and 6 gives the following results.

Fig. 5-A.

Beam No. 1 $\frac{2500 \times 16 \times 21 \times 28}{20 \times 7 \times 21} =$ 8000 lbs.

Beam No. 2 $\frac{2500 \times 16 \times 28}{20 \times 7} =$ 8000 lbs.

Beam No. 3 $\frac{2500 \times 36 \times 10 \times 28}{20 \times 22\frac{1}{2} \times 7} =$ 8000 lbs.

Beam No. 4 $\frac{2500 \times 36 \times 10 \times 21 \times 28}{20 \times 22\frac{1}{2} \times 7 \times 21} =$ 8000 lbs.

Fig. 5-B.

Beam No. 1 $\frac{2500 \times 12 \times 24}{24 \times 6} =$ 5000 lbs.

Beam No. 2 $\frac{2500 \times 12 \times 18 \times 24}{24 \times 6 \times 18} =$ 5000 lbs.

Beam No. 3 $\frac{2500 \times 36 \times 10 \times 18 \times 24}{24 \times 30 \times 6 \times 18} =$ 5000 lbs.

Beam No. 4 $\frac{2500 \times 36 \times 10 \times 24}{24 \times 30 \times 6} =$ 5000 lbs.

Fig. 6-A.

Beam No. 1 $\frac{4000 \times 15 \times 22\frac{1}{2} \times 18 \times 24}{20 \times 30 \times 6 \times 18} =$ 9000 lbs.

Beam No. 2 $\frac{4000 \times 15 \times 22\frac{1}{2} \times 24}{20 \times 30 \times 6} =$ 9000 lbs.

Beam No. 3 $\frac{4000 \times 35 \times 12 \times 22\frac{1}{2} \times 24}{20 \times 28 \times 30 \times 6} = 9000$ lbs.

Beam No. 4 $\frac{4000 \times 35 \times 12 \times 22\frac{1}{2} \times 18 \times 24}{20 \times 28 \times 30 \times 6 \times 18} = 9000$ lbs.

Fig. 6-B.

Beam No. 1 $\frac{1400 \times 15 \times 21}{21 \times 7} = 3000$ lbs.

Beam No. 2 $\frac{1400 \times 15 \times 28 \times 21}{21 \times 7 \times 28} = 3000$ lbs.

Beam No. 3 $\frac{1400 \times 36 \times 10 \times 28 \times 21}{21 \times 24 \times 7 \times 28} = 3000$ lbs.

Beam No. 4 $\frac{1400 \times 36 \times 10 \times 21}{21 \times 24 \times 7} = 3000$ lbs.

The above illustrations show that it is a very simple matter to calculate the braking force on any car regardless of the system of levers.

RAIL AND BRAKE SHOE FRICTION.

Having fixed upon the method of calculating brake pressures, the next problem is to analyze the relation of the friction between the wheel and rail and retardation caused by brake application. This relationship is fixed by the coefficient of friction between wheel and rail and between wheel and shoe. By coefficient of friction is meant the ratio of the weight of an object to the force required to slide it on a level surface, thus:

Assume a block of iron weighing 10 lbs. on a plane surface, a cord being attached to the block passing over a pulley attached to a scale pan, as indicated in Fig. 8. Supposing it requires 2½ lbs. in the scale pan to cause the block of iron to slide, the ratio of this amount to the weight of the block is called the coefficient of friction. It is evident that since 2½ is one-fourth of 10 lbs., the coefficient of friction in this case will be 25%.

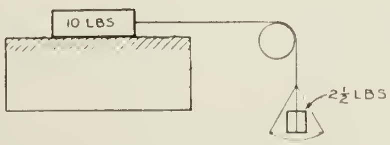


FIG. No. 8.

Friction Between Wheel and Rail.

Many experiments have been made to ascertain the coefficient of friction between wheel and rail, in other words, the force required to slide a loaded wheel. One series of tests were made by Geo. L. Fowler of the Schoen Steel Wheel Co., as reported in book entitled "THE CAR WHEEL." These results are as follows:

Load on Wheel	*Chilled Iron Wheel*	*Steel Wheel*
2000 lbs	28.7	28.5
4000	25.9	25.4
6000	25.4	24.5
8000	24.2	24.6
10000	23.3	23.8
12000	22.3	23.7
16000	21.9	23.2
20000	22.0	23.6
24000	22.4	23.5
28000	21.7	23.6
30000	21.4	23.4

TABLE NO. 2A

It will be noted from the above coefficients of friction that they are not particularly different for the chilled iron wheel as compared with the steel wheel, the particular item brought out in each series of tests is that the coefficient of friction decreases as the load increases. Similar tests were made by the Association of Manufacturers of Chilled Car Wheels at Purdue University for various conditions of rail; also for four different loads on the wheel, namely, 2808 lbs., 6840 lbs., 12000 lbs. and 20000 lbs. In the first test, the railhead was planed to correspond with the taper of the wheel. The results in this case were as follows:

RAIL HEAD PLANED DOWN TO FIT TAPER OF WHEEL

PRESSURE BETWEEN WHEEL AND RAIL	CHILLED WHEEL TANG. PULL	CHILLED WHEEL COEF. OF FRICTION	STEEL WHEEL TANG. PULL	STEEL WHEEL COEF. OF FRICTION
2808 lbs.	796	28.4	558	19.9
6840	1724	25.2	1461	21.3
12000	2757	23.0	2841	23.7
20000	3312	16.6	3533	17.7

NOTE:—Bearing areas on the rail are shown in Fig. 9.

TABLE NO. 3.

These figures agree very well with those obtained by Mr. Fowler, except that there is a greater falling off after the load of 12000 lbs. is reached.

Another test was made with a normal rail, except that it was flooded with engine oil, the wheels having the regular M. C. B. tread with 1 in 20 cone. The results were as follows:

RAIL HEAD AS ROLLED -- WHEEL AND RAIL
FLOODED WITH ENGINE OIL

PRESSURE BETWEEN WHEEL AND RAIL	CHILLED WHEEL TANG. PULL	CHILLED WHEEL COEF. OF FRICTION	STEEL WHEEL TANG. PULL	STEEL WHEEL COEF. OF FRICTION
2808 lbs.	537	19.1	557	19.8
6840	1198	17.5	1173	17.1
12000	2289	19.1	2116	17.6
20000	2890	14.5	2910	14.5

TABLE NO. 4.

STEEL WHEEL.

CHILLED IRON WHEEL.

Fig. No. 9.

Condition of rail after slipping of chilled iron and steel wheel under pressures of 2,808, 6,840, 12,000 and 20,000 pounds, rail having first been planed to a taper of 1 in 20 to fit the standard M. B. C. tread contour.

Purdue University Friction Tests.

STEEL WHEEL
AREAS
1—.097″ 2—.140″ 3—.165″ 4—.340″

CHILLED IRON WHEEL
AREAS
1—.070″ 2—.125″ 3—.160″ 4—.220″

FIG. No. 10.
Condition of rail after slippage of chilled iron and steel wheel under loads of 2,808, 6,840, 12,000 and 20,000 pounds respectively.
Purdue University Friction Tests.

As in the previous cases, there was very little difference shown in the tangential pull in either type of wheel, the coefficient was much higher, however, than would be expected under the conditions of rail. However, the loads were so heavy that the oil was practically pressed out from between the irregularities in the surface and, therefore, allowing the surface of the wheel to come in contact with the surface of the rail and still allow a frictional contact, whereas on lubricated surfaces with lighter loads the oil forms a film between the two rubbing surfaces so that they do not come in contact.

On a third test, a normal rail was used with the regular M. C. B. 1 in 20 cone, the following results being obtained:

RAIL HEAD AS ROLLED

PRESSURE BETWEEN WHEEL AND RAIL	CHILLED WHEEL TANG. PULL	CHILLED WHEEL COEF. OF FRICTION	STEEL WHEEL TANG. PULL	STEEL WHEEL COEF. OF FRICTION
2808 lbs.	818	29.1	1127	40.0
6840	1737	25.4	2093	30.5
12000	2688	22.4	2863	23.8
20000	3286	16.4	3432	17.1

NOTE:—Effect on the rail is shown in Fig. 10.

TABLE NO. 5.

We may assume that under ordinary wheel loads encountered in railway service we may secure coefficients of friction between the wheel and rail as follows:

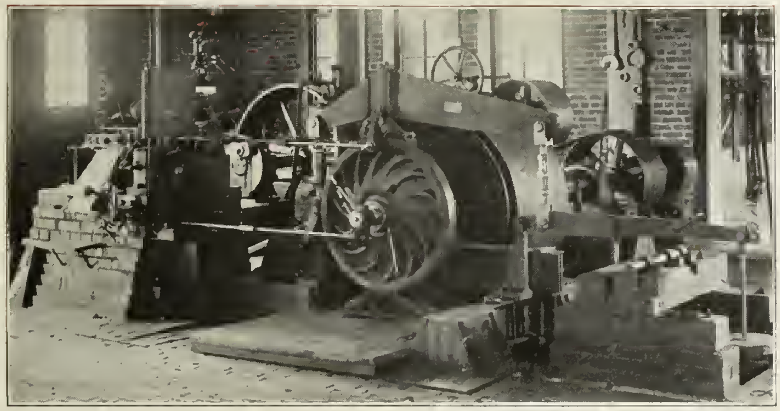
On a clean dry rail ....20 to 25 per cent
Clean thoroughly wet rail ....18 to 20 per cent
Oily and moist rail ....15 to 18 per cent
Sleet on rail ....15 per cent
Light snow or frost on rail ....10 per cent

In each case, the use of sand will increase the coefficient of friction about 50%. These figures are sufficiently accurate for ordinary calculations, having to do with friction between wheel and rail.

Brake Shoe Friction.

The friction between a moving wheel and the brake shoe is a variable quantity, depending upon many conditions, such as amount of shoe pressure, length of time shoe is applied, kind of shoe, kind of insert, condition of shoe, etc. The M. C. B. Association for the past 18 years has carried on a large number of tests on both chilled iron and steel wheels at Purdue University and also at Mahwah, N. J., to determine the coefficient of friction under widely varying operating conditions.

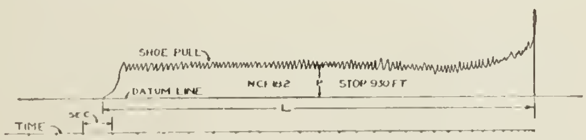
The brake shoe testing machine used, is so arranged that a wheel may be mounted upon a shaft, upon which is also mounted a heavy fly wheel to represent the equivalent energy of a moving car. The shaft is actuated either by steam engine or electric motor. The shoe is applied as shown in Fig. 11, at the top of the wheel, the pressure being delivered through a system of levers, so arranged that any desired pressure may be secured by the addition of weights. The tangential pull is secured by a crosshead and connecting rod joined to the dynamometer shown at the left. A graphic record is obtained on the chart, which shows the pull of the brake shoe, the distance traveled and time consumed.



Method of determining coefficient of friction against spinning between wheel and rail. Purdue University Friction Tests.

FIG. No. 11.

Fig. 12 shows a record taken from the brake shoe machine. The pull of the brake shoe is indicated by the height of the sawtooth line above the base line. It will be noted that this pull remains fairly constant for three-quarters of the distance required to make the stop, after which it increases until just at the point of stopping it increases to double the average amount. There are two opposite tendencies at work during the progress of the stop which tend to equalize each other, —first, the coefficient of friction has a tendency to rise as the speed is reduced; second, the coefficient of friction is lowered as the temperature of the shoe increases. For this reason, in making a stop, the coefficient of friction is about uniform until near the end of the stop when a very material increase occurs. The mean coefficient of friction is found by taking the area of the card with a planimeter and dividing this by the length, which gives the average coefficient.



Chronograph Record from Brake Shoe Testing Machine.

FIG. No. 12.

The quantities in Table 6 show the coefficients obtained by the M. C. B. Association for various pressures above 2800 lbs. The object of all these tests was for the purpose of studying the relation of brake shoe pressures to stopping a car. These quantities are of little importance in relation to controlling trains on grades because much smaller pressures are required which produce higher coefficients of friction and less heating of the shoe and wheel, and less loss of brake shoe metal per unit of work accomplished.

Shoe Number	Laboratory at which test was made	Brake Shoe	Mean coefficient in per cent initial speed of 40 m. p. h. chilled iron wheel. Shoe pressure, lbs. 2808	4152	6840	Mean coefficient in per cent stops from an initial speed of 65 m.p.h. Steel tired wheel. Shoe pressure, lbs. 2808	4152	6840	10000	12000	15000	18000
281	A. B. S. & F. Co.	Plain cast iron, (C. & N. W.)	26.3	21.7	21.	16.3	13.1	11.0	....	....	....	....
282	Purdue	Plain cast iron, (C. & N. W.)	22.1	21.6	20.4	16.	....	12.4	....	10.4	....	....
283	A. B. S. & F. Co.	Plain cast iron (without re-inf.)	25.1	23.5	20.6	....	....	11.7	....	....	....	....
284	Purdue	Plain cast iron (without re-inf.)	30.3	27.7	24.5	16.3	....	13.5	....	11.6	....	....
285	A. B. S. & F. Co.	Congdon, seven wrought inserts	26.8	19.0	15.3	19.7	17.7	12.4	8.9	9.4	8.2	7.7
286	Purdue	Congdon, seven wrought inserts	22.2	19.8	16.4	20.6	....	14.0	....	11.3	....	....
287	A. B. S. & F. Co.	Congdon, five wrought inserts	25.0	18.3	17.2	20.3	18.0	11.8	9.5	9.8	8.5	7.6
288	Purdue	Congdon, five wrought inserts	24.4	22.6	19.1	15.1	....	11.9	....	11.7	....	....
289	A. B. S. & F. Co.	Streeter, two hard iron inserts	24.5	22.6	19.0	16.9	14.9	11.2	11.7	10.4	9.5	9.1
290	Purdue	Streeter, two hard iron inserts	21.3	20.6	16.4	13.6	....	10.8	....	10.7	....	....
291	A. B. S. & F. Co.	Lappin, chilled ends (A. B. S. & F.)	18.2	16.8	16.1	15.0	13.4	10.1	8.8	8.6	8.8	6.8
292	Purdue	Lappin, chilled ends (A. B. S. & F.)	20.5	19.6	18.9	17.0	....	13.0	....	11.1	....	....
293	A. B. S. & F. Co.	Lappin, chilled ends (A. B. S.)	20.5	18.4	14.3	16.3	15.1	11.6	9.1	9.3	7.9	6.6
294	Purdue	Lappin, chilled ends (A. B. S.)	18.4	17.8	17.5	16.9	....	12.7	....	12.2	....	....
295	A. B. S. & F. Co.	Plain cast iron (A. B. S. Co.)	27.0	25.1	21.9	16.9	13.5	11.3	9.7	8.4	9.3	8.5
296	Purdue	Plain cast iron (A. B. S. & Co.)	21.0	20.3	18.5	16.2	....	13.2	....	11.1	....	....
297	A. B. S. & F. Co.	Plain cast iron (Columbia B. S. Co.)	27.0	28.6	21.8	18.3	14.0	13.5	15.3	....	....	....
298	Purdue	Plain cast iron (Columbia B. S. Co.)	21.0	18.9	17.3	16.8	....	13.1	....	10.7	....	....
299	A. B. S. & F. Co.	Diamond S, chilled ends (A.B.S.Co.)	24.2	20.0	16.2	21.5	17.4	13.5	11.2	10.8	9.8	9.8
300	Purdue	Diamond S, chilled ends (A.B.S.Co.)	22.8	20.5	18.3	17.3	....	13.6	....	12.3	....	....
301	A. B. S. & F. Co.	Walsh, two hard iron inserts	22.6	20.0	14.9	14.7	12.1	10.3	8.7	8.6	9.1	8.7
302	Purdue	Walsh, two hard iron inserts	23.7	20.5	19.8	16.6	....	14.4	....	11.5	....	....
303	A. B. S. & F. Co.	Pittsburg, malleable iron shell	24.4	21.9	17.0	17.7	17.9	17.5	14.0	11.8	11.2	10.7
304	Purdue	Pittsburg, malleable iron shell	26.8	25.4	21.5	22.8	....	18.9	....	17.6	....	....
305	A. B. S. & F. Co.	Pittsburg, steel shell (P.B.S.Co.)	29.9	29.6	24.2	23.0	20.9	18.7	15.8	14.7	14.2	15.3
306	Purdue	Pittsburg, steel shell (P.B.S.Co.)	29.4	27.5	23.4	25.8	....	23.2	....	22.2	....	....
307	A. B. S. & F. Co.	National, chilled ends (N.B.C.Co)	16.3	15.2	11.9	15.1	11.3	9.8	8.2	7.5	6.9	8.3
308	Purdue	National, chilled ends (N.B.C.Co.)	19.3	16.4	14.3	15.2	....	12.1	....	11.2	....	....
		Averages	23.6	21.4	18.5	17.7	15.3	13.3	10.9	11.4	9.4	9.0

TABLE NO. 6.

In reviewing the foregoing table we note that there is a variation of more than 50% under the same operating condition, due to difference in shoe and character of wheel; also that as the shoe pressure increases, the coefficient of friction decreases; that as the velocity in creases, the coefficient of friction decreases.

From the above tests, the M. C. B. Association has at different times fixed their specifications for brake shoes requiring the following standards of coefficient of friction:

FIRST: Tests upon chilled iron wheel from an initial speed of 40 M. P. H.

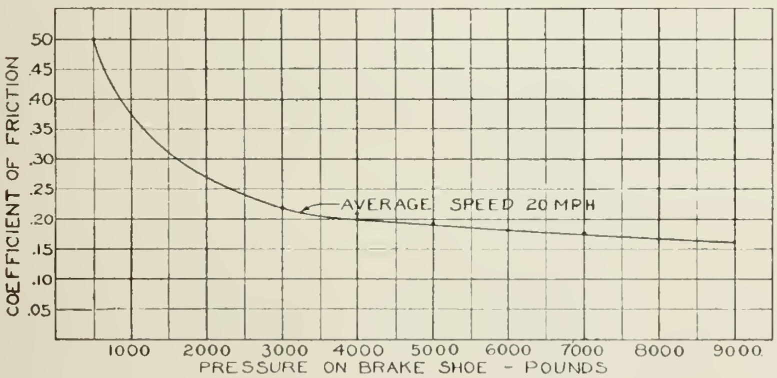
at 2808	lbs. pressure	22%
at 4152	" "	20%
at 6840	" "	16%

SECOND: Tests upon steel wheel from an initial speed of 65 M. P. H.

at 2808	lbs. pressure	16%
at 4152	" "	14%
at 6840	" "	12½%
at 12000	" "	11%

It is self evident that a review of the tests indicate a higher coefficient of friction for chilled iron than for steel wheels, and this feature is remarkably prominent in the M. C. B. specification, for, after making allowance for the decreased coefficient of friction on account of stops made from 65 M. P. H. on the steel wheel as compared with 40 M. P. H. on the chilled iron wheel, which may amount to as much as 15%, there is still a difference of 15% to 20% in the coefficient of friction in favor of the chilled iron wheel.

Fig. 13 indicates about the coefficient of friction that may be expected on chilled iron wheels.



COEFFICIENT OF FRICTION
DIAMOND 'S' BRAKE SHOE ON CHILLED IRON WHEEL
Fig. No. 13.

WORK OF ACCELERATION AND RETARDATION.

The work done by the engine in excess of that required to overcome train resistance manifests itself in increasing velocity. This is called acceleration and acts as a storage for surplus energy which is given up again by assisting the locomotive in climbing grades, coasting with steam shut off, or when stops or reduced speed are required, the brake shoe is used to transform the stored energy into heat. The loss

of energy manifests itself in a retarded motion until a full stop is reached. In addition to the transformation of stored energy in the train, the brakes may be called upon to offset the effect of gravity on descending grades.

Every moving body acquires a momentum or force due to the motion of the body. The magnitude of this force, which may be designated as energy of motion, increases directly as the weight of the body increases and as the square of its velocity. In mechanics the formula for energy of motion is expressed as follows:

$$E = \frac{W V^2}{64.32}$$

in which E equals the stored energy and W equals the weight of the moving body, and V the velocity in feet per second.

When applied to movement of cars, the velocity is always expressed in miles per hour, which introduced into the above formula, and reducing to lowest terms becomes:

$$\frac{2000 \times \left(\frac{5280}{3600}\right)^2}{64.32} = 66.887783 \text{ ft. lbs. of energy in 1 ton at 1 M. P. H.}$$

From the above formula tables 8 and 9 have been prepared, showing the energy stored for various loads at various velocities:

TABLE OF VELOCITIES.

Miles per Hour	Feet per Second	Feet per Minute	Miles per Hour	Feet per Second	Feet pe Minute
1	1.467	88	10	14.667	880
2	2.933	176	15	22.000	1320
3	4.400	264	20	29.333	1760
4	5.867	352	25	36.667	2200
5	7.333	440	30	44.000	2640
6	8.800	528	35	51.333	3080
7	10.267	616	40	58.667	3520
8	11.733	704	45	66.000	3960
9	13.200	792	50	73.333	4400
			60	88.000	5280

TABLE NO. 7.

SHOWING THE EQUIVALENT VELOCITIES IN MILES PER HOUR, FEET PER SECOND AND FEET PER MINUTE.

ENERGY OF MOTION IN FOOT POUNDS
VARIOUS LOADS AT VARIOUS VELOCITIES
VELOCITY IN M. P. H.

Load in tons	5	10	15	20	25	30	35	40
1	1672	6689	15050	26755	41805	60199	81938	107020
2	3344	13377	30099	53510	83610	120398	163875	214041
3	5017	20066	45149	80265	125415	180597	245813	321061
4	6689	26755	60199	107020	167219	240796	327750	428082
5	8361	33444	75249	133776	209024	300995	409688	535102
10	16722	66888	150498	267551	418049	601990	819375	1070205
15	25083	100332	225746	401327	627073	902985	1229063	1605307
20	33444	133776	300995	535102	836097	1203980	1638751	2140409

TABLE NO. 8.

ENERGY OF MOTION IN FOOT POUNDS
M. C. B. GROSS LOADS AT VARIOUS VELOCITIES

Capacities	30 ton	40 ton	50 ton	70 ton
Gross Load:				
Pounds	95000	132000	161000	210000
Tons	47.5	66.0	80.5	105.0
Velocity M. P. H.				
5	79429	110365	134612	175580
10	317717	441459	538447	702322
15	714863	993284	1211505	1580224
20	1270868	1765837	2153787	2809287
25	1985732	2759121	3365292	4389511
30	2859453	3973134	4846020	6320896
35	3892033	5407877	6595971	8603441
40	5083472	7063350	8615146	11237148

TABLE NO. 9.

ENERGY IN ROTATING WHEELS.

The wheels under a car have two motions, first, the forward motion, the same as any other part of the car, the moving energy of which is the same, pound for pound, as any other part of the car; and in addition to this is the second motion, which consists of rotating about the center of the axle.

It is evident that additional energy is required to produce this motion. The law which governs this case is exactly the same as for a body moving in a straight line, and the only complication is that no two points on a radial line have the same velocity of rotation; the parts near the center of the axle are moving in a very small circle, while those at the circumference move in a large circle, hence have a greater velocity. To overcome this difficulty, a point is found at which, if all the metal of the wheel and axle were concentrated, the result would be the same when the metal is distributed as in the wheel and axle. This point is found for various classes of wheels to vary from 8¾" to 9" from the center of the wheel, and for the purpose of calculation, 8⅞" has been shown to represent all wheels.

Since the velocity of the train is measured on the circumference of the wheel having a radius of 16½", the average velocity of the axle and wheel, due to rotation, will be $\frac{8.875}{16\frac{1}{2}}$ which, substituted in the formula for moving energy becomes:

$$\left(\frac{8.875}{16.5}\right)^2 x \frac{66.888}{2000} = .0096766 \left\{\begin{array}{l} x\ 6900(60M) = 66.76854 \\ x\ 8000(80M) = 77.41280 \\ x\ 8900(100M) = 86.12174 \\ x10500(140M) = 101.60430 \end{array}\right\} \begin{array}{l}\text{Ft. lbs. rota-} \\ \text{ting energy in} \\ \text{whls. under 1} \\ \text{car at 1 MPH}\end{array}$$

from which the following table is calculated, showing the energy of rotation for various velocities for the 60,000 lb., 80,000 lb. 100,000 lb. and 140,000 lb. capacity cars:

ENERGY IN ROTATING WHEELS—FOOT POUNDS—
STANDARD CARS

Capacity	60000 lbs.	80000 lbs.	100000 lbs.	140000 lbs.
Wgt. of 8 Whls. & Axles	6900 lbs.	8000 lbs.	8900 lbs.	10500 lbs.
1 M. P. H.	67	77	86	102
5	1669	1935	2153	2540
10	6677	7741	8612	10160
15	15023	17418	19377	22861
20	26707	30965	34449	40642
25	41730	48383	53826	63503
30	60092	69672	77510	91444
35	81791	94831	105499	124465
40	106830	123860	137795	162569
Ratio of Rotating Energy in wheels to Total Energy in car—approximately	2.10%	1.75%	1.60%	1.45%

TABLE NO. 10.

There are two sources from which the energy represented by the quantities in the above tables can be acquired, namely, gravity and motive power, and the rate at which either of these can store up energy in the form of motion is determined by the amount of force remaining after deducting the amount necessary to overcome journal and rolling friction, curve resistance, ascending grade resistance, etc., in other words, the force of gravity on descending grades or the drawbar pull originating from the motive power is exactly balanced or equalized by journal or rolling friction, curve resistance, ascending grade resistance, and energy stored in the form of velocity. Each of these items is susceptible of independent analysis.

The items which enter into the retardation of a train are:

First—Train resistance.

Second—Curve resistance.

Third—Gravity resistance.

Fourth—Brake resistance.

TRAIN RESISTANCE..

The term "train resistance" is intended to cover all elements such as journal and rolling friction, wind resistance, velocity resistance, etc., which occur on straight level tracks.

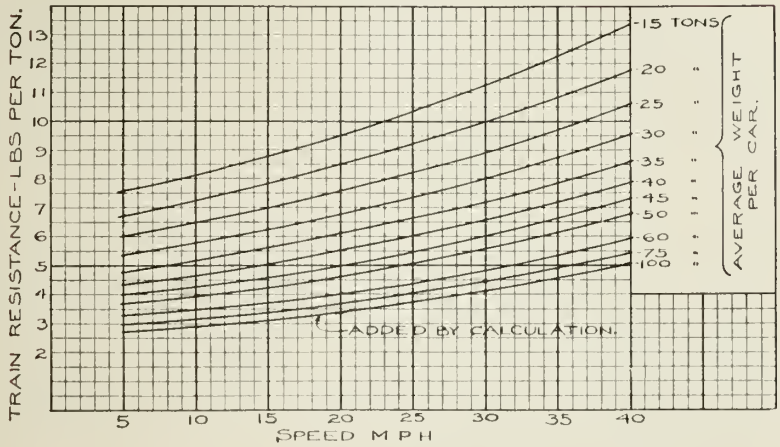
One of the most recent and conclusive tests to determine the amount of this resistance under varying conditions of load and velocity was conducted on the Illinois Central R. R. under the direction of Edw. C. Schmidt, Professor of Railway Engineering, University of Illinois. The results are contained in University of Illinois Bulletin No. 43. The amount of resistance per ton established in these tests is shown in the following table, No. 11, and shown graphically in Fig. 14. The quantity for 100 ton cars is not given in these tests, but is estimated from the graphical diagram.

TRAIN RESISTANCE — LBS. PER TON
WEIGHTS PER CAR — TONS

MPH	15	20	25	30	35	40	45	50	55	60	75	*100
5	7.6	6.8	6.0	5.4	4.8	4.4	4.0	3.7	3.5	3.3	3.0	2.7
10	8.2	7.3	6.5	5.8	5.2	4.7	4.3	4.0	3.7	3.5	3.2	2.9
15	8.8	7.9	7.0	6.3	5.6	5.1	4.6	4.2	3.9	3.7	3.4	3.1
20	9.6	8.5	7.6	6.8	6.1	5.5	5.0	4.6	4.3	4.0	3.7	3.4
25	10.4	9.3	8.3	7.4	6.7	6.0	5.5	5.0	4.7	4.4	4.0	3.7
30	11.3	10.0	9.0	8.0	7.3	6.6	6.0	5.5	5.1	4.9	4.5	4.2
35	12.3	10.9	9.7	8.8	7.9	7.2	6.6	6.1	5.7	5.4	4.9	4.6
40	13.4	11.8	10.6	9.5	8.6	7.9	7.3	6.8	6.3	6.0	5.5	5.1

NOTE:* Estimated.

TABLE NO. 11.



THE RELATION BETWEEN RESISTANCE AND SPEED
FOR VARIOUS AVERAGE WEIGHTS OF CARS

FIG. No. 14.

CURVE RESISTANCE.

Curve resistance arises from lateral slippage of wheels arising from continual change in direction and longitudinal slippage due to difference in length of inside and outside rail and in addition to this, the grinding of the flange against the side of the head of the rail is an element of varying amount, being relatively small in new flanges but increasing very rapidly as the flange wears away and approaches a vertical condition.

The amount of tread slippage per degree of curve per mile of track is as follows:

1st front wheel	6.788	feet
2nd " "	5.068	"
1st rear wheel	4.515	"
2nd " "	0	
	16.371	feet
Average of all wheels	4.093	"

In addition to this there is a grinding of the forward flange of each truck, the amount of slippage depending upon the amount to which the flange is worn. If the side bearing is ⅛" below the tread bearing there will be a slippage amounting to 40 feet per mile, but if the point of bearing should be ¼" below the rail, the slippage will be 80 feet per mile. We may assume for an average condition that the side bearing of the flange is ⅛" below the top of the rail, in which case the slippage will be 40 feet per mile, and since this occurs on but one wheel in four, the average per wheel will be 10 feet per mile, and since the pressure of the flange against the rail is ¾ of the load carried by the wheel, the slippage for a full load would amount to ¾ of 10 or 7.5 feet which, added to the 4 feet for the tread slippage would make a total of 11.5 feet per mile. Assuming the coefficient of friction to be .25 it would require 500 lbs. to slide one ton. The total curve resistance then would be represented by:

$$\frac{500 \times 11.5}{5280} = 1.09 \text{ lbs.}$$

It is evident that the curve resistance is a varying quantity, depending upon the condition of the wheels, and may vary from ⅜ lbs. per ton per degree of curve for tread slippage alone, to 1¾ lbs. per ton per degree of curve when flanges are badly worn.

For the purposes of this paper we will use ¾ lbs. per ton per degree of curve as an average quantity, which is shown in the following graphical diagram, Fig. 15.

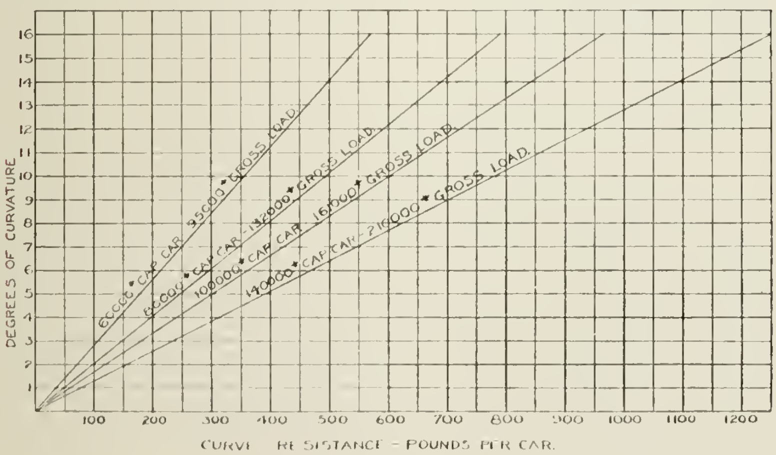


FIG No. 15.

GRAVITY RESISTANCE.

The effect of gravity for various loads on various grades is shown in Table No. 12. This table indicates the pull in lbs. per ton on grades of various percentages. The pull for gross load of 30, 40, 50 and 70 ton cars and also the energy that would be developed in one mile of 1, 2, 3 and 4% grades respectively, is shown in the same table.

GRAVITY EFFECT FOR VARIOUS LOADS ON VARIOUS GRADES.

Load in Tons.	Pull in lbs. grade. 1%	2%	3%	4%	Ft. lbs. developed in one mile grade. 1%	2%	3%	4%
1	20	40	60	80	105600	211200	316800	422400
2	40	80	120	160	211200	422400	633600	844800
3	60	120	180	240	316800	633600	950400	1267200
4	80	160	240	320	422400	844800	1267200	1689600
5	100	200	300	400	528000	1056000	1584000	2112000
10	200	400	600	800	1056000	2112000	3168000	4224000
15	300	600	900	1200	1584000	3168000	4752000	6336000
20	400	800	1200	1600	2112000	4224000	6336000	8448000
M. C. B. Gross Loads:								
47.5	950	1900	2850	3800	5016000	10032000	15048000	20064000
66.0	1320	2640	3960	5280	6969600	13939200	20908800	27878400
80.5	1610	3220	4830	6440	8500800	17001600	25502400	34003200
105.0	2100	4200	6300	8400	11088000	22176000	33264000	44352000

TABLE NO. 12.

BRAKE RESISTANCE.

The tangential pull of the brake shoe in lbs. of retarding effort for various coefficients of friction is shown in Table No. 13.

The quantities shown in bold face type indicate the normal tangential pull that may be expected from the pressure shown at the head of the column for speeds of 20 M. P. H. and correspond with the coefficient of friction shown immediately under the pressure at the head of the column. These coefficients indicate what may normally be expected on chilled iron wheels from the respective pressures:

RETARDING EFFORT OF BRAKE SHOES IN LBS. FOR VARIOUS COEFFICIENTS OF FRICTION.

Coef. of Fric.	PRESSURE ON BRAKE SHOE IN LBS. AND AVGE. COEFFICIENT FOR THAT PRESSURE 500	700	1000	2000	3000	4000	5000	7500	10000	15000	20000
	50%	43%	36%	27%	23%	20%	18%	14%	12½	10%	9%
5%	25	35	50	100	150	200	250	375	500	750	1000
10%	50	70	100	200	300	400	500	750	1000	**1500**	**2000**
15%	75	105	150	300	450	600	750	**1125**	**1500**	2250	3000
20%	100	140	200	400	600	**800**	**1000**	1500	2000	3000	4000
25%	125	175	250	**500**	**750**	1000	1250	1875	2500	3750	5000
30%	150	210	300	600	900	1200	1500	2250	3000	4500	6000
35%	175	245	**350**	700	1050	1400	1750	2625	3500	5250	7000
40%	200	280	400	800	1200	1600	2000	3000	4000	6000	8000
45%	225	**315**	450	900	1350	1800	2250	3375	4500	6750	9000
50%	**250**	350	500	1000	1500	2000	2500	3750	5000	7500	10000

TABLE NO. 13.

PROBLEMS IN ACCELERATION.

While the subject of acceleration is somewhat foreign to this discussion, yet it seems pertinent to illustrate the fundamentals of these problems which represent the force exerted to produce motion in cars, all of which must be accounted for in train resistance, curve and grade resistance or energy stored in motion. The foregoing tables contain all the necessary data.

EXAMPLE NO. 1.

What velocity will be acquired by a train of 50 loaded 100,000 lbs. capacity cars two miles from the starting point, on straight level track, when pulled by an engine having 200,000 lbs. on its drivers, assuming the coefficient of friction to be 20%?

We have the following data:

Drawbar pull—200,000x20%.................. 40,000 lbs.
Wgt. of train 50x80.5........4025 tons
Wgt. of engine .... 180 tons

Total wgt. of train .... 4,205 tons
Train resistance at 6 lbs. per ton.............25,230 lbs.
Distance traveled 2 miles....................10,560 feet
Input—drawbar pull multiplied by distance:
40,000 x 10,560..................422,400,000 ft. lbs.
Energy used in train resistance:
25,230 x 10,560 ....266,428,800 ft. lbs.

Difference = energy of motion.........155,971,200 ft. lbs.
Rotating energy of wheels approx. 1.6% 2,495,539 ft. lbs.

Difference, energy in train velocity ....153,475,661 ft. lbs.

Energy of velocity per ton $\frac{153,475,661}{4205}$ = 36,498 ft. lbs.

Referring to table No. 8 we find this corresponds to a velocity of between 20 and 25 miles. The exact velocity is found by the formula taken from Table No. 8.

$$\sqrt{\frac{36,498}{1672}} \times 5 = 23.4 \text{ miles.}$$

Thus the theoretical velocity is 23.4 miles, but some experience is necessary for close calculations, taking into consideration the kind of weather, temperature, winds, etc., but the above are the main elements in the problem.

EXAMPLE NO. 2.

How many loaded 100,000 lb. capacity cars (80.5 tons gross load) can an engine weighing 180 tons and having 200,000 lbs. on its drivers, start on a 1% grade and acquire a velocity of 20 miles per hour in 15 minutes:

Drawbar pull 200,000 lbs. x .20 = 40,000 lbs.
Average velocity $\frac{20}{2}$ = 10 M. H. P.
Distance traveled—15x880 ft. (Table No. 7) 13,200 ft.
Gross load of car..............................80.5 tons
Acquired energy of motion at 20 MPH
(Table No. 9) ....2,153,787 ft. lbs.
Energy of rotation (Table No. 10)...... 34,449 ft. lbs.
Total energy per car...................2,188,236 ft. lbs.

Train and curve resistance 80.5 x 6 lbs. = 483 lbs.
Gravity effect per car (Table No. 12) =1610 lbs.
2093 lbs.

Total retarding effect 2093 x 13,200 = 27,627,600 ft. lbs.
Total input 40,000 x 13,200 = 528,000,000 ft. lbs.
Total consumption of energy:
Energy of motion ....2,188,236 ft. lbs.
Energy due to gravtiy ....27,627,600 ft. lbs.
Total ....29,815,836 ft. lbs.

The number of cars that can be handled will then be:

$$\frac{528,000,000}{29,815,836} = 17.7$$

Deducting 2¼ cars for engine equivalent leaves 15 cars. This indicates that under the conditions of the problem, 15 cars can be handled on a 1% grade and accelerate to the rate of 20 miles per hour in 15 minutes.

EXAMPLE NO. 3.

A train of 30 loaded 100,000 lb. capacity cars (gross load 80.5 T.) starts at the foot of a 1% grade at a speed of 35 MPH. At what point on the grade will it stop, assuming the engine to be working under full head of steam.

Weight of train:
30 cars 80.5 tons =........2415 tons
Locomotive and tender = 180 tons (D. B. pull 40,000 lbs.)
Total ....2595 tons

Energy at foot of grade at 35 MPH.
In velocity 2595 T.x81938 (Table 8)=212,629,110 ft. lbs.
In rotation of whs. (Table 10) 1.60% = 3,402,066 ft. lbs.
Total energy ....216,031,176 ft. lbs.

Amount of energy destroyed per foot:
By gravity effect 1% of 2595 T. (5,190,000 lbs.)=51,900 lbs.
By train resistance 2595 T. at 4 lbs. per ton=10,380 lbs. (Table No. 11)
By curve resistance—assume ¼ of the distance consists of 4° curves — Avge. is then

$$\frac{2595 \times 3}{4} \text{ .... } = 1946 \text{ lbs.}$$

Total ....64,226 lbs.
Less drawbar pull of locomotive ....40,000 lbs.
Loss of energy per foot ....24,226 lbs.

Distance traveled before coming to stop:

$$\frac{216,031,176}{24,226} = 8917 \text{ feet or } 1.69 \text{ miles}$$

Time required to travel this distance: ....

$$\frac{60 \text{ min.}}{17.5 \text{ (Avg. speed)}} \times 1.69 = 5.8 \text{ min. or } 5 \text{ min. } 48 \text{ sec.}$$

The above problems are sufficient to show in general the relation between drawbar pull, velocity, grade, etc., but before problems of this kind can be handled practically, it is necessary to have considerable experience regarding the effect of winds, condition of rail, etc. The problems given are only intended to illustrate in a general way how such problems are solved.

IRREGULARITIES IN BRAKING POWER OF FREIGHT CARS.

VARIATIONS IN PRESENT STANDARDS.

It has long been known that there is a gross irregularity in the braking power of various classes of cars having the same gross load but no association or committee has ever compiled the entire freight equipment of the United States to ascertain to what extent these variations occur.

Knowing that any attempt to intelligently discuss the many problems growing out of the braking power of cars, without having access to this information, would be impossible, the author took it upon himself to request from all the railroads and private lines of the United States detailed information relative to the tare weight of their entire equipment. The response was most gratifying, as prompt replies gave the desired information for 1,900,000 cars. This represents such a large percentage of the total that it furnishes an entirely satisfactory basis for any general problem having to do with standards affecting the retardation of trains.

FREIGHT CAR EQUIPMENT
SUMMARY OF TARE WEIGHTS OF FREIGHT CARS.

No. of Cars	Average Weight	Braking Power at 60%	Braking Power % of Gross Load	Retarding Force at 20% Coef. of Friction	Equivalent to Gravity Effect * on Grade of Per cent. Indicated -
RAILROAD CARS 60,000 LBS. CAP.					
81	12074	7200	7.6	1440	1.9
430	18053	10800	11.4	2160	2.7
19312	22955	13800	14.5	2760	3.3
79261	27458	16500	17.4	3300	3.9
261206	31871	19100	20.1	3820	4.4
103728	36231	21700	22.8	4340	5.0
17593	41240	24700	26.0	4940	5.6
7737	45993	27600	29.1	5520	6.2
1076	51867	31100	32.8	6220	6.9
Total 490424					
Average	32317	19400	20.4	3880	4.5
RAILROAD CARS, 80000 LBS. CAP.					
33	23697	14200	10.8	2840	2.5
22215	27862	16700	12.7	3340	2.9
118597	32296	19400	14.7	3880	3.3
242076	36583	21900	16.6	4380	3.7
130451	41476	24900	18.9	4980	4.1
14043	46310	27800	21.1	5560	4.6
2720	50219	30100	22.8	6020	4.9
104	57942	34800	26.4	6960	5.6
Total 530239					
Average	36793	22100	16.8	4420	3.7
RAILROAD CARS 100,000 LBS. CAP.					
2257	28945	17400	10.8	3480	2.5
40273	32451	19500	12.1	3900	2.8
216012	37819	22700	14.1	4540	3.2
202743	42077	25200	15.6	5040	3.5
90126	46118	27700	17.2	5540	3.8
6072	50955	30600	19.0	6120	4.1
20	56000	33600	20.9	6720	4.5
Total 557503					
Average	40429	24300	15.1	4860	3.4

(Continued on page 34.)

(Continued from page 33.)

RAILROAD CARS 140,000 LBS. CAP.

4497	52889	31700	15.1	6340	3.3
10	60000	36000	17.1	7200	3.7
Total 4507					
Average	52905	31700	15.1	6340	3.3

*.4% grade deducted for train resistance.

TABLE NO. 14.

SUMMARY OF TARE WEIGHTS OF CARS OF PRIVATE CAR LINES 60000 LBS. CAPACITY.

No. of Cars	Average Weight	Braking Power at 60%	Braking Power % of Gross Load	Retarding Force at 20% Coef. of Friction	Equivalent to Gravity Effect on Grade of % Indicated*
904	29000	17400	18.3	3480	4.1
17929	33241	19900	20.9	3980	4.6
30772	36817	22100	23.3	4420	5.1
17867	42490	25500	26.9	5100	5.8
25833	47077	28200	29.7	5640	6.3
6237	51899	31100	32.8	6220	6.9
199	59000	35400	37.3	7080	7.9
Total 99741					
Average	40765	24500	25.8	4900	5.6

*.4% grade deducted for train resistance.

TABLE NO. 14a.

IRREGULARITIES IN APPLICATION OF PRESENT STANDARDS.

Wherever the braking ratios of cars have been examined for any purpose, the same result is always found, namely, considerable irregularity in the percentage of braking power in various classes of cars.

To show about what may be expected, a number of cars were measured at random, and the results are indicated in Table No. 15. This shows that we may expect a variation of from 40% to 80% in the braking power of freight cars. That in case of refrigerator cars, we may rather expect a higher braking power where a 10" cylinder is used than where an 8" cylinder is used, and that the braking power is quite frequently in excess of the M. C. B. recommended standard. Experience indicates that there is a very serious lack of uniformity in the percentage of braking power and that brake rigging is practically never checked in order to correct these irregularities. This seems to be a case where "if ignorance is bliss, 'tis folly to be wise." The condition, however, is far from satisfactory from any standpoint. The recommended braking power for cars is certainly one in which wide liberties are taken, so much so that the exceptions are almost as numerous as the cases where the rule is followed.

For Table See Page 36 and 37.

IRREGULARITIES IN RETARDATION.

Not only is the deviation from the standard rule a serious proposition, but a glance at the summary of equipment for various capacities of cars will indicate that the rule itself creates a very serious situation with reference to the braking power of loaded cars.

According to the M. C. B. rule, the retardation for all empty cars, regardless of the tare weight, is uniform, therefore, each car, regardless of its character, will have a tendency to come to a standstill in exactly the same distance as any other car when the brakes are applied. But when the cars are loaded, a very different condition prevails, for in some instances the braking power is less than 15% of the load, where-

as in other cases it is more than 30%. This gives rise to severe shocks in a train of miscellaneous cars, especially if the cars of each of these extreme classes are grouped together.

For example—assume 20 refrigerator cars of the private car lines, weighing 55,000 lbs. tare, 95,000 lbs. gross, and 20 gondola cars of 100,000 lbs. capacity, 40,000 lbs. tare and 161,000 lbs. gross:

Refrigerator Cars:

Braking power 60% of 55,000 lbs.	=33,000 lbs.
Retarding force 20% of 33,000 lbs.	= 6,600 lbs.

Gondola Cars:

Braking power 60% of 40,000 lbs.	=24,000 lbs.
Retarding force 20% of 24,000 lbs.	= 4,800 lbs.

Retardation refrigerator cars	20x6,600 lbs.=	132,000 lbs.
Retardation gondola cars	20x4,800 lbs.=	96,000 lbs.

Total retardation 40 cars ....228,000 lbs.

Average retardation per car ....5,700 lbs.

Energy at 40 M. P. H. (See Table No. 9):

20 ref. cars=20x5,083,472 ft. lbs.=101,669,440 ft. lbs.

20 gon. cars=20x8,615,146 ft. lbs.=172,302,920 ft. lbs.

273,972,360 ft. lbs.

Number of feet required to stop the train (ignoring train resistance)

$$\frac{273{,}972{,}360 \text{ ft. lbs.}}{228{,}000 \text{ lbs.}} = 1201.63 \text{ ft.}$$

Refrigerator cars would stop, if alone, (ignoring train resistance) in:

$$\frac{101{,}669{,}440 \text{ ft. lbs.}}{132{,}000 \text{ lbs.}} = 770 \text{ feet}$$

Gondola cars would stop, if alone, (ignoring train resistance) in:

$$\frac{172{,}302{,}920 \text{ ft. lbs.}}{96{,}000 \text{ lbs.}} = 1795 \text{ feet.}$$

The average stop is 1201.63 feet, which, divided into the energy in the refrigerator cars will give the retarding force used per foot in stopping the refrigerator cars, thus:

$$\frac{101{,}669{,}449 \text{ ft. lbs.}}{1201.63 \text{ feet}} = 84609 \text{ lbs.}$$

Similarly, the retarding force used per foot to stop the gondola cars, independent of the refrigerator cars is:

$$\frac{172{,}302{,}920 \text{ ft. lbs.}}{1201.63 \text{ ft.}} = 143391 \text{ lbs.}$$

Since the effective retarding force of the 20 refrigerator cars is 20x6600 lbs. or 132000 lbs., the refrigerator cars are destroying, in addition to the energy contained in themselves, a considerable portion of the energy in the gondola cars, which is the difference between 132,-000 lbs. and 84609 lbs., or 47,391 lbs., which represents the pull on the drawbar between the two groups of cars, assuming, of course, that the 100,000 lb. cap. gondola cars are ahead and the refrigerator cars are behind.

TABLE SHOWING VARIATIONS IN LENGTHS OF BRAKE LEVERS UNDER INDIVIDUAL CARS SELECTED OF RANDOM

Type of Car	Tare Wgt. of Car	Dia. of Cylinder	CYLINDER LEVER Receiving End Beams No. 1 and 2	CYLINDER LEVER Receiving End Beams No. 3 and 4	CYLINDER LEVER Delivering end All Beams	INTERMED. LEVER Receiving End	INTERMED. LEVER Delivering End (Full Length of Lever)	SECOND INTERMEDIATE LEVER See Fig. 6 A) Receiving End	SECOND INTERMEDIATE LEVER Delivering End (Full Length of Lever)	FLOATING TRUCK LEVER Receiving End Inside Beam	FLOATING TRUCK LEVER Receiving End Outside Beam	FLOATING TRUCK LEVER Delivering End Both Beams	FIXED TRUCK LEVER Receiving End (Full Length of Lever)	FIXED TRUCK LEVER Delivering End	*TOTAL BRAKING POWER	PER CENT OF BRAKING POWER
STYLE OF BRAKE RIGGING SHOWN IN FIG. 5-A.																
Illustration Fig. 5	.....	8"	16	36	20	10	22½			28	21	7	28	21	32000 lbs.	....
Ref.	54300	10	14½	44	29½	6	18½			24	18	6	24	18	30600 "	56.4
"	54300	10	14½	39½	25	10½	29			28	21	7	26	19½	34900 "	64.3
"	53900	10	10½	30½	20	10½	30½			24	18	6	24	18	33000 "	61.3
"	53500	10	14½	39½	25	10¾	29½			28	21	7	26	19½	35800 "	67.0
"	53500	10	14½	43	28½	6¼	18			24½	18½	6	24	18	33200 "	62.0
"	52100	10	15	37½	22½	8	21			28	19	9	25	17	31900 "	61.2
"	50200	10	5	23½	18½	4	12			22½	16½	6	22½	16½	20400 "	40.7
"	49400	10	13	39	26	7	21			26	19½	6½	28	21	30600 "	62.0
"	46700	10	10½	31½	21	10	31			24	18	6	24	18	30900 "	66.2
"	46400	8	14½	35½	21	10½	25½			32	24	8	27	20	27100 "	58.4
"	45200	8	14½	36	21½	10½	25½			28	21	7	25¼	19	26500 "	58.6
"	45100	10	12½	31	18½	11½	29½			25	17	8	25	18	32100 "	71.2
"	44600	10	11½	30	18½	9	28			25	17	8	25	18	28000 "	62.8
"	44300	10	10½	31½	21	10½	31			24	18	6	24	17¾	31600 "	71.3
"	43300	10	12	33	21	8½	22½			24	16	8	24	16	30200 "	69.8
"	41700	8	11¾	28¾	17	11½	27½			31½	22½	9	31½	22½	25100 "	60.3
Furn.	45900	10	14	33½	19½	10¾	26¼			25½	17	8½	25½	17	33400 "	72.8
Box	42400	10	10	31	21	10	31			24	18	6	24	18	29900 "	70.5
"	42300	10	9	31½	22½	9½	31½			24	18	6	24	18	25800 "	61.0
"	40800	10	10	32	22	9½	31½			27	19	8	24	18	23700 "	58.0
"	38800	8	13	34½	21½	8½	22			25¼	18	7¼	25	18	21200 "	54.6
"	38700	10	10½	35	24½	6½	21½			28	21	7	26	19½	27200 "	70.3

"	38600	8	13½	34½	21	5½	14				30	21½	8½	30	21½	22600	"	58.5
"	37900	8	13½	34	20½	5½	14				30	21½	8½	30	21½	23900	"	63.1
"	36100	10	9¾	29¾	20	6¼	19				27	19	8	27	19	26200	"	72.6
"	32300	8	13	37	24	10½	29				32	24	8	28	21	21800	"	67.5
Stock	30300	8	12	34	22	12	33				28	20	8	28	20	19000	"	62.7
Poultry	43500	8	13	27¼	14¼	8	18				24	18	6	24	18	35800	"	82.3
STYLE OF BRAKE RIGGING SHOWN IN FIG. 5-B.																		
Illustration Fig. 5	.....	8"	12	36	24	10	30				18	24	6	24	18	20000	"	....
Furn.	38100	8	11	32	21	8½	24½				17	22½	5½	22½	17	20400	"	53.5
Gond.	37000	8	14½	31	16½	9½	21½				19½	28	8½	28	19½	28200	"	76.3
STYLE OF BRAKE RIGGING SHOWN IN FIG. 6-A.																		
Illustration Fig. 6	.....	10"	15	35	20	12	28	22½	30		24	18	6	24	18	36000	"	....
Ref.	62700	10	20	40	20	20	40	20	30¼		26½	19	7½	26½	19	36800	"	58.7
"	62100	10	15½	30	14½	15½	30	17	30		26½	19	7½	26½	19	33500	"	54.0
"	61200	10	19¼	39¼	20	19½	40	17¾	30¼		26½	19	7½	26½	19	31300	"	51.2
"	60100	10	16	31	15	15½	30	17	30		26½	18½	8	26½	18½	31400	"	52.2
"	58300	10	15½	30½	15	15½	30	17	30		26½	18½	8	26½	19	30500	"	52.3
Box.	52500	10	21½	40½	19	21½	40½	17	30		26½	19	7½	26½	19	36800	"	70.0
"	48100	10	18	40½	22½	17½	40	20	30		26	18½	7½	26	18½	29200	"	60.8
"	46000	8	17	30	13	16½	30	18	30		26½	18½	8	26½	18½	25600	"	55.7
"	44700	10	15	30	15	15	30	15	30		26½	18½	8	26½	18½	26100	"	58.5
"	44700	10	14½	30	15½	16	30½	15½	30		26½	19	7½	26½	19	27300	"	61.1
"	39600	8	17½	30	12½	13½	31	17½	30		28½	20	8½	28½	20	23200	"	58.6
Gond.	49300	10	18	37½	19½	17	36	18	30½		24½	17½	7	24½	17½	23100	"	46.9
"	39300	10	14¾	28¾	14	17½	36½	12	30		26	18½	7½	26	18½	22200	"	56.5
STYLE OF BRAKE RIGGING SHOWN IN FIG. 6-B.																		
Illustration Fig. 6	.....	6	15	36	21	10	24				28	21	7	21	28	12000	"	....
Ref.	46800	8	15	32½	[illegible]	9½	20¾				19¼	14	5¼	12	16¼	23000	"	49.1
"	46300	8	16½	18	34½	11½	24				19	14	5	12	16	26800	"	57.9
"	.....	8	12	33	21	12	27				19	14	5	12	16	18000	"	....

NOTE: * The total braking power shown differs slightly, in many cases, from the product of the levers shown, which is due to small irregularities in the lengths of some levers under the same car.

TABLE NO. 15.

The same result will be obtained if we use the retarding effect on the gondola cars as a basis. Here the energy to be destroyed per foot is .... 143991 lbs.
and the effective retarding force is 20x4800 lbs. or.......... 96000 lbs.

The difference represents the measure of retarding force per foot obtained from the refrigerator group, which is transmitted through the drawbar between the two groups of cars, viz........ 47391 lbs.

Should the refrigerator cars be placed ahead and the gondola cars behind, the action between the two groups will be one of compression or buckling, instead of a pull. The force of the action would, however, be the same in both cases.

The magnitude of the force acting between two groups of cars, due to dissimilar braking ratios, can be obtained by a much shorter method than the one outlined. As an illustration, we will use the example given above:

	Gross Load	Retarding Force	Per cent of Retarding Force to Gross Load
One 60000 lb. cap. Ref. car	95000 lbs.	6600 lbs.	6.9474
One 100000 lb. cap. Gond. car	161000 lbs.	4800 lbs.	2.9814
Totals	256000 lbs.	11400 lbs.	
Average	128000 lbs.	5700 lbs.	4.4531

The percentages shown indicate that for every 100 lbs. of gross load in the refrigerator car we have a retarding force of 6.9474 lbs., and the same relation in the 100000 lbs. car and average. Therefore, for every 100 lbs. of gross load in the refrigerator car we have an excess retarding force of the difference between 6.9474 lbs. and 4.4531 lbs. or 2.4943 lbs. The total surplus retarding force of one car is then 950x 2.4943 lbs. or 2369.5 lbs. For 20 cars this force becomes 20x2369.5 lbs. or 47390 lbs., which is the same result as obtained under the first method.

This can also be verified by using the gondola car as a basis for our deductions. We then have a deficiency in braking the car per 100 lbs. of gross load, which is made up from the surplus in each refrigerator car, of the difference between 4.4531 lbs. and 2.9814 lbs. or 1.4717 lbs. This multiplied by 1610 (the number of hundred weight in the gross load of the gondola car) gives a deficiency in retarding force in each gondola car of 2369.5 lbs. or for 20 cars a total deficiency of 47,390 lbs.

In the example used we have assumed all of the cars to be loaded to full capacity. We will now consider the shock in a train travelling at a speed of 40 M. P. H., consisting of 20, 100,000 lb. capacity gondola cars loaded to capacity, and 20 empty, 60,000 lb. capacity refrigerator cars, weighing light 55,000 lbs. each, when given a full service application.

The relative figures are then:

	Gross Load	Retarding Force	Per cent Retarding Force to Gross Load
One 60000 lb. cap. Refrigerator	55000	6600	12.0000
One 100,000 lb. cap. Gondola	161000	4800	2.9814
Totals	216000	11400	....
Average	108000	5700	5.2777

Excess retarding force per 100 lbs. of gross load of the refrigerator car is 12.0000—5.2777 or	6.7223 lbs.
The total per car is 550x6.7223 lbs. or	3697.2 lbs.
Total excess for 20 cars is 20x3697.2 lbs. or	73,944 lbs.
Deficiency in retarding force per 100 lbs. of gross load of the 100,000 lb. cap. gondola car is 5.2777—2.9814 or	2.2963 lbs.
The total per car is 1610x2.2963 lbs. or	3697.2 lbs.
Total deficiency for 20 cars is 20x3697.2 lbs. or	73,944 lbs.

To afford a full appreciation of the magnitude of these shocks, we will compare them with the drawbar pull of locomotives in average service.

Assume two locomotives of the following proportions:

	MIKADO	MALLET COMPOUND
Weight on drivers	200,000 lbs.	400,000 lbs.
Coef. of friction between drivers and rail	20%	20%
Max. drawbar pull is then	40,000 lbs.	80,000 lbs.

Comparing these figures, which represent the maximum drawbar pull that could be delivered by these locomotives, we find an alarming situation. The shock of 47390 lbs. between the groups of 20 loaded refrigerator cars and 20 loaded gondola cars is seen to be greater than could possibly be delivered by the Mikado locomotive, while the shock between the 20 empty refrigerator cars and the 20 loaded gondola cars (73,944 lbs.) very nearly equals the maximum pull that could be delivered by the Mallet Compound locomotive. Or, comparing another way—the shock in the first instance is equal to that which would be produced by 20, 100,000 lb. cap. cars with air brakes fully applied if placed in the same train with 20 cars without air brakes; or, the pull of 73,944 lbs. would be the equivalent of 30 cars with air brakes fully applied joined to 30 cars without air brakes.

Is it any wonder that draft gears are pulled apart, and cars are buckled when shocks of this magnitude can so readily occur? But this is not the most serious consequence of irregularity in braking power. We will now study into the heat developed by the brake shoes.

HEAT DEVELOPED BY BRAKE SHOE FRICTION.

In showing the relation of excessive continuous braking power to wheel service, the most important item is heat, and therefore, it is well to reduce the element of heat to terms that are easily grasped.

The office of the brake shoe is to transform the surplus energy in a moving car into heat on the tread of the wheel. There is a definite relation between mechanical energy and heat which makes the calculation of the amount of heat easy.

Experimenters along this line have found that approximately 775 foot pounds of energy will raise the temperature of one pound of water one degree Fahr. This quantity is called the British Thermal Unit. In studying iron, however, it is found that it will increase in temperature at a much faster rate than water, the ratio being eight to one, the specific heat of iron being approximately .125. This means that to heat 1 lb. of iron one degree will require $\frac{775}{8}$ foot lbs., or say, for the purpose of easy calculation 100 foot lbs., and since 2000 degrees may be taken as the temperature at which iron melts, it will require 2000x 100 or 200,000 foot lbs. to melt one lb. of iron. In the following dis-

cussion, we will use the amount of heat required to melt a pound of iron on the tread of the wheel as the unit for comparison.

Table No. 16 shows the heat developed by brakes on the tread of a wheel represented in pounds of melted iron that would contain the equivalent amount of heat in the average 30, 40, 50 and 70 ton car per minute if the brakes were fully applied and the speed of the car 30 miles per hour. The table also shows the heat equivalent in pounds of melted iron required to offset the effect of gravity on 30, 40, 50 and 70 ton cars on grades of 1, 2, 3 and 4% respectively.

There are, however, few grades of any considerable length on any trunk line above 2%. The table would indicate that for any car operating at a uniform velocity of 30 miles per hour on a long down grade, that the heat developed on the tread would not exceed the equivalent heat required for melting from 3 to 6 lbs. of iron per minute, depending upon whether the capacity of the car is 30 tons or 70 tons.

EQUIVALENT HEAT IN LBS. OF MELTED IRON GENERATED ON THE TREAD OF ONE WHEEL PER MINUTE UNDER CONDITIONS SHOWN.

	Where Full Braking Power is Used at 30 MPH		To Maintain a Constant Speed of 30 MPH on the Respective Grades			
Capacity	Avge.	Max.	1%	2%	3%	4%
30 Ton R. R.	6.4	11.0	1.1	2.7	4.2	5.8
30 Ton Priv.	8.1					
40 Ton	7.3	10.0	1.5	3.7	5.9	8.1
50 Ton	8.0	10.5	1.9	4.5	7.2	9.8
70 Ton	10.5	10.9	2.4	5.9	9.3	12.8

NOTE—Train resistance deducted—6 lbs. per ton.

TABLE NO. 16.

Under the column showing the heat developed by the full average braking power of the car it is noted that in the 30 ton private car line class, the possibility of developing heat is practically the same as in the 50 ton class, and, therefore, when the private car line cars are used in miscellaneous trains, the heat developed on the tread of the wheel is as great or greater on the 30 ton private car line car than that developed on the tread of the wheel carrying the 50 ton car. Also, if the 30 ton private car is equipped with an 8" cylinder, the brake leverage is greater and the shoe travel smaller, requiring a fine adjustment to maintain an 8" piston travel, and where this additional inspection is not given to the private cars, a short piston travel will be more frequent than under the heavier cars, so that it is entirely possible on grades of 1% to have a full braking power of the private car develop, when only 25% of the braking power of the 50 ton cars is required. Furthermore, if we take the heaviest car in the 60,000 lb. class, which weighs 55,000 lbs., the full normal braking power at 30 miles per hour would be the equivalent of the heat required to melt 10.9 lbs. of iron per minute, which is greater than that on the average 70 ton car, and if this car should chance to have a brake hanger 30 degrees from normal, this quantity might be increased 20%, representing the heat equivalent of over 13 lbs. of melted iron per minute; whereas only one-fifth of this amount is required for the ordinary grades over which the car would pass. This indicates a condition which not only produces shocks in the train, bends the brake beams, causes an excessive consumption of brake shoes, but also leaves its effect upon the tread of the wheel.

IRREGULARITIES IN BRAKE SHOE APPLICATION ON DESCENDING GRADES.

The work required of brakes is two-fold:

1st: To retard the velocity of a train by using up the stored energy, causing the train to slow down or stop.

2nd: To maintain a uniform velocity on descending grades by producing a retarding force just equal to the gravity effect.

The manner in which a heavy train is handled on a long descending grade has a marked influence on the braking power required which, in turn, is the measure by which heat is generated on the tread of the wheel.

If the brakes are applied early and a uniform velocity maintained, there will be a minimum braking power required, whereas if the velocity is allowed to increase to a maximum before brake application is made, the maximum braking power is necessary at a reduced coefficient of friction.

To illustrate. Assume a train of 70 ton cars loaded, descending a 2% grade, 10 miles long. There are two methods of operating the brakes on long continuous grades:

1st: Intermittent application.

2nd: Continuous application.

It is self evident that in either case the total retardation for the whole distance is alike, but in the first case the brakes are applied heavily to check the speed of the train and then released to allow the speed to pick up. When this method is employed a speed of 40 M. P. H. may be acquired at the top of the grade before service application of the brake is made. The velocity is reduced to 20 M. P. H. in 45 seconds. The average velocity for the whole distance is 30 M. P. H.

When the second method is employed a light service application is made as soon as the train acquires 25 or 30 M. P. H., and is held as regularly as possible, giving the train a uniform velocity of 30 M. P. H. so that the time of descent is the same in either case.

There is a vast difference, however, in the effect on brake rigging, brake shoe consumption and heated wheels, as can readily be shown by the following analysis:

FIRST METHOD.

In the analysis of this problem we will assume a single car, as the train is simply made up of a number of units of the same kind. The items entering into the problem are as follows:

Gross weight of car	105 tons
Tare weight of car	60000 lbs.
Braking power	40000 lbs.
Grade effect (See Table 12)	
2% of 210000 lbs	4200 lbs.
Journal and rolling friction	
4¼ lbs. per ton (See Table 11)	446 lbs.
Curve resistance, say 4°	
⅔ of distance or 2⅔° the entire distance	210 lbs.
(See Fig. No. 15).	
Stored energy at 20MPH (Table No. 9)	= 2,809,287 ft. lbs.
Energy in rotating wheels (Table No. 10)	= 40,642 ft. lbs.
	2,849,929 ft. lbs.

Stored energy at 30 MPH = 6,320,896 ft. lbs.
Energy in rotating wheels = 91.444 ft. lbs.

6,412,340 ft. lbs.

Stored energy at 40 MPH = 11,237,148 ft. lbs.
Energy in rotating wheels = 162,569 ft. lbs.

11,399,717 ft. lbs.

Assuming the train to have a velocity of 20 M. P. H. at the time the full train started on the down grade, the distance traveled before acquiring a velocity of 40 M. P. H. is calculated:

Energy per car at 40 M. P. H. = 11,399,717 ft. lbs.
Energy per car at 20 M. P. H. = 2,849,929 ft. lbs.

Difference equals energy acquired by grade effect ....= 8,549,788 ft. lbs.
Grade effect on one car.............. 4200 lbs.
Less—Journal and rolling friction....446
Curve resistance ....210 656 lbs.

Net pull on each car due to gravity........3544 lbs.
Number of feet traveled in increasing velocity from

20 M. P. H. to 40 M. P. H. $= \dfrac{8{,}549{,}788}{3544 \text{ lbs.}} = 2412$ ft.

Average velocity is 30 M. P. H.
Time required to travel 2412 feet at 30 M. P. H.
(See Table No. 7) $= \dfrac{2412}{44} = 55$ sec.

According to the problem, the brakes are now applied to check the velocity to 20 M. P. H. in 45 seconds. The energy to be destroyed by the brakes in 45 seconds amounts to:

Distance traveled in 45 sec. at 30 MPH = 44 ft. x 45 = 1980 ft.
Grade effect = 3544 lbs. x 1980 = 7,017,120 ft lbs.
Plus difference in energy between
40 M. P. H. and 20 M. P. H. 8,549,788 ft. lbs.

Total energy to be destroyed by brakes in 45 seconds ....15,566,908 ft. lbs.

Energy destroyed per second $= \dfrac{15{,}566{,}908}{45} = 345{,}931$ ft. lbs.

Energy destroyed per foot $= \dfrac{15{,}566.908}{1980} = 7862$ ft. lbs.

Brake pressure required coefficient of friction 20% = 7862 x 5 = 39310 lbs.
Pressure per shoe $= \dfrac{39310}{8}$ = 4914 lbs.

This represents the full braking power of the car.

After checking the velocity to 20 M. P. H., the cycle is completed, a complete cycle being:

2412 ft. acceleration—time 55 sec.
1980 ft. deceleration—time 45 sec.

4392 ft. cycle time 100 sec.

Total distance = 10 miles = 52800 feet
Number of cycles $\frac{52800}{4392}$ = 12 cycles plus 96 feet

Time required to complete 12 cycles
12 x100 sec. = 1200 sec. or 20 min.

Total energy absorbed in the 12 cycles:
15,566,908 ft. lbs. x 12 = 186,802,896 ft. lbs.

SECOND METHOD.

Assume as before that the train started down the grade at 20 MPH with the intention of running at a uniform speed of 30 MPH over the entire 10 miles. The distance required for increasing the velocity from 20 MPH to 30 MPH is:

Energy at 30 MPH =	6412340 ft. lbs.
Energy at 20 MPH =	2849929 ft. lbs.
Diff. equals energy acquired by grade effect	3562411 ft. lbs.
Number of feet traveled is	$\frac{3562411}{3544}$ = 1005 ft.

Time required to travel 1005 feet (average speed 25 MPH or 36.67 ft. per second) = $\frac{1005}{36.67}$ = 27 seconds

At this point the brakes are applied and because of the lower pressure required, the coefficient of friction rapidly increases to say, 25%. The braking pressure required to just equal the net grade effect would be 3544 x 4 = 14176 lbs.

Pressure per shoe = $\frac{14176}{8}$ = 1772 lbs.

This is slightly over one-third the braking power of the car.

Time required to descend grade:

1st 1005 feet	27 sec.
Remainder, 51795 ft. at 44 ft. per sec.	1177 sec.
Total	1204 sec. or 20 min. 4 sec.

The point must not be lost sight of in the above illustration that where the speed is intermittent, the full braking power of the car is required, and that the braking power used in this illustration is greater than used under the average 70 ton car; that the heat developed per minute is equivalent to the heat required to melt 13 pounds of iron which, in turn, causes the maximum temperature on the tread of the wheel, causing the most violent expansion of the metal, placing the greatest strain on all parts of the brake rigging, great shocks in the train; consuming double the amount of brake shoe metal per unit of work done on account of the extreme heating of the brake shoe, and taking the maximum chance for a runaway, with nothing whatever gained, because experience has conclusively demonstrated that a smaller, continuous brake application as shown by the second method is safe; allows the same average speed; requires about one-third of the brake pressure, but because of the greater coefficient of friction between brake shoe and wheel, the work accomplished is nearly one-half of that which is secured with three times the pressure. The equivalent heat per minute is represented by 5.8 lbs. of metal melted, which certainly shows conclusively the material advantage as regards heating the tread of the

wheel. To obtain a uniform velocity requires the use of retainers, which are extremely advantageous on all heavy grades, although the majority of cars, perhaps, which are operated on comparatively level tracks do not have the retainer equipment maintained in working order. The retainer equipment, however, on difficult grades is of prime importance.

From no standpoint can the heavy intermittent pressure be justified as compared with the continuous brake application at a much lower pressure. The intermittent application of brakes is similar to what would occur on a level track if an engine should start out with a full head of steam and sanded rail for a minute or two and then shut off all steam and coast for a minute, and again apply the steam, continuing to use steam and coasting in cycles of two or three minutes. It would be just as absurd as to operate a train on a down grade at a variable speed, applying and releasing brakes in cycles of two and three minutes. This item will be referred to later under the subject of—temperature stresses in the body of the wheel.

IRREGULARITIES CAUSED BY ANGULARITY OF BRAKE HANGER.

The inequality of brake pressures is often aggravated by the position of the brake hanger. Figs. 16 and 17 show two radically different angles of brake beam suspension. When the brake hanger occupies a position at right angles to a line drawn from the center of the wheel to the center of the brake shoe, the stress in the hanger will be at a minimum and equal to the tangential pull of the brake shoe which represents the tendency of the shoe to rise or fall according to the direction of rotation of the wheel. A hanger in this position has no effect whatever on the pressure of the shoe against the wheel. On the other hand, with the hanger in the position shown in Fig. 16, the short support to the brake beam acts as a toggle, and in case the wheel is rotating in a direction to cause the shoe to rise, a very material increase is given to the brake shoe pressure which, in turn, increases the compression in the hanger. Whereas, on the other pair of wheels in the same truck, where the pull is downward, the action is the reverse, and the pressure is diminished; for that reason the tension in the hanger is reduced. To show how much this action amounts to the following analysis is given:

Let A equal forward wheel

B equal rear wheel

P equal shoe pressure on wheel

S equal stress in hanger

a equal angle of hanger with M. C. B. normal position

P′ equal brake pressure from cylinder

f equal coefficient of friction

$$S = \frac{fP}{\cos a}$$

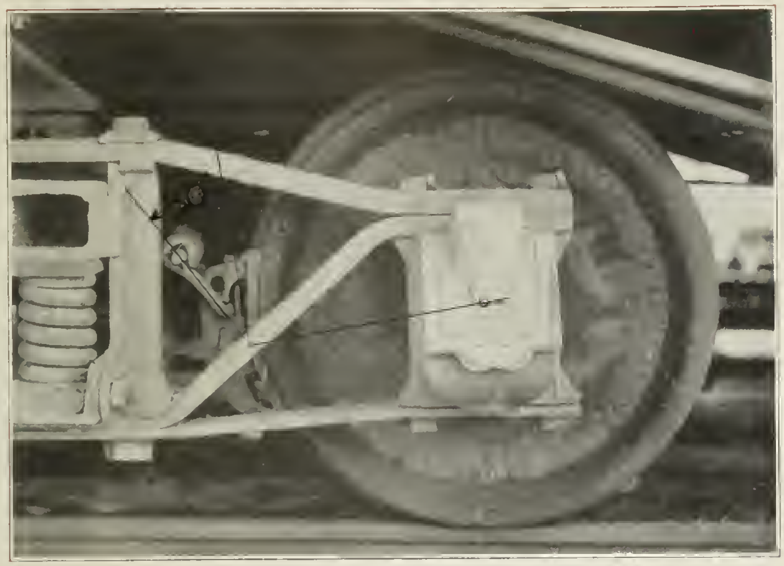
The final increase or decrease in the effective braking power will be determined by the following equation:

P—P′ equals P f tan a

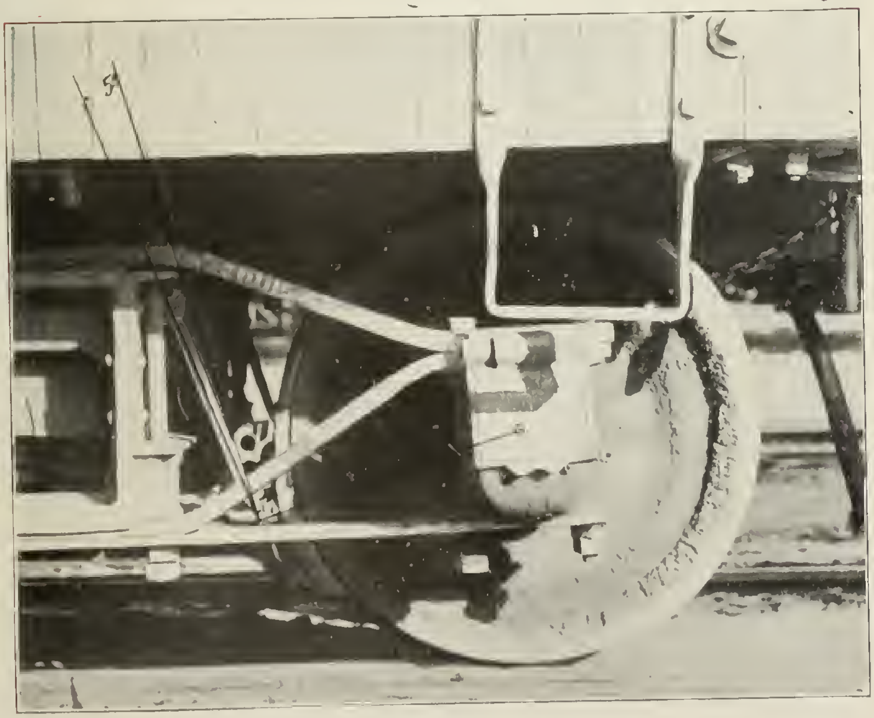
Then $\frac{P'}{P} = 1$ plus (or minus) f tan a

Or P equals $\frac{P'}{1 \text{ plus (or minus) } f \tan a}$

plus being used for the forward wheel and the minus for the rear wheel.



Improper position of brake hanger.
FIG. No. 16.



Proper position of brake hanger
FIG. No. 17.

The coefficient of friction (f) varies with the velocity and ranges from 25% for low speed to 10% for high pressures at a speed of 30 miles per hour.

From the formula the curves shown in Fig. 18 are plotted and show the percentage of effective braking power that will be obtained for every degree of angle that the brake shoe hanger is off from the Master Car Builders' normal position. From these curves it is seen that there is no advantage to be obtained by placing the brake shoe support at an angle. Although the braking force will be increased on one pair of wheels, it will be correspondingly decreased on the other pair, and the result, in all probability, will be flat spots, brake burns on the wheels having excessive braking power.

Fig. 19 shows the effect of a short toggle brake hanger which on account of the excessive compression was bent laterally and drove the brake beam endwise so that the entire braking force was obtained on one flange, the brake shoe overlapping the rim of the mate wheel, which resulted in a very unequal wearing of the brake shoe as shown in the sketch, all of the wear taking place on the side of the shoe so that when one side of the shoe had worn out, the shoe was reversed and the other side worn out, without decreasing the thickness of the shoe. Fig. 20 shows samples of misplaced brake shoes which add very materially to the pressure per sq. in. on the contact between brake shoe and wheel, which in turn very materially increase the loss of brake shoe metal and increases the local heating of the tread of the wheel.

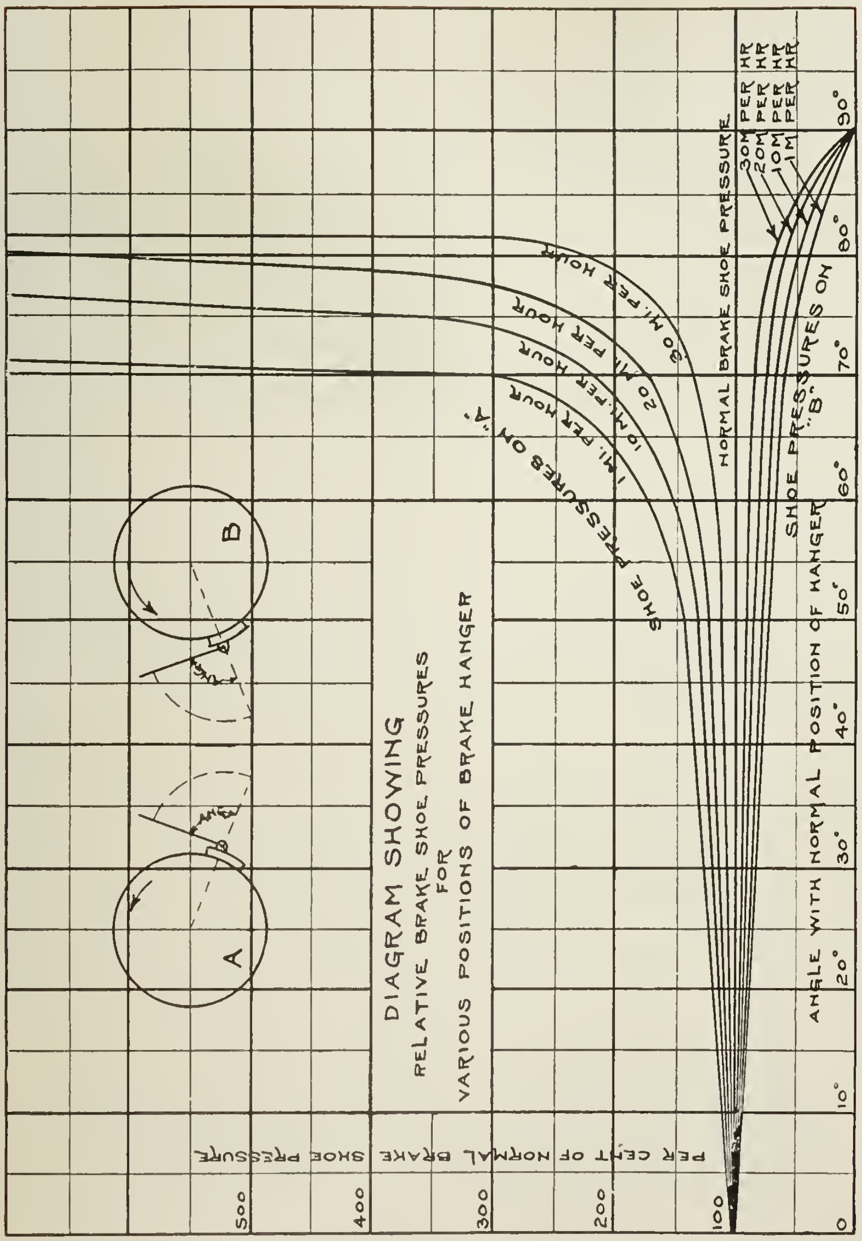
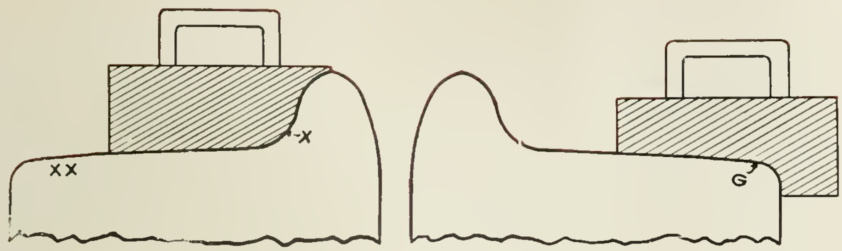
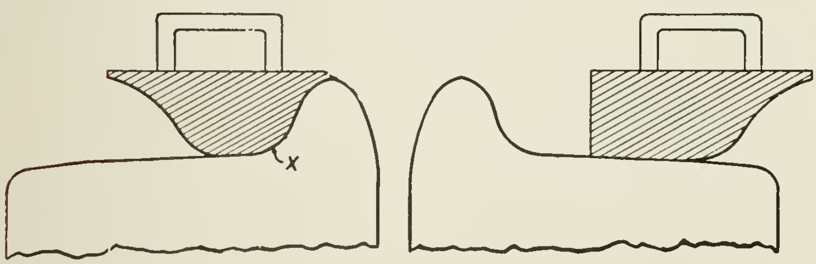


DIAGRAM SHOWING
RELATIVE BRAKE SHOE PRESSURES
FOR
VARIOUS POSITIONS OF BRAKE HANGER
A
B
PER CENT OF NORMAL BRAKE SHOE PRESSURE
100
200
300
400
500
ANGLE WITH NORMAL POSITION OF HANGER
0
10°
20°
30°
40°
50°
60°
70°
80°
90°
NORMAL BRAKE SHOE PRESSURE
SHOE PRESSURES ON "A"
1 MI. PER HOUR
10 MI. PER HOUR
20 MI. PER HOUR
30 MI. PER HOUR
SHOE PRESSURES ON "B"
30M PER HR
20M PER HR
10M PER HR
1M PER HR

Fig. No. 18.



Sketch showing start of one brake shoe to bear against the flange while the other shoe bears on rim. In these cases the points "X" and "G" show blue temper and the space from "X" to "XX" shows no heating.



Sketch showing brake shoe worn in flange and then turned around to finish wearing. This shows one shoe again bearing at the throat of wheel, causing blue temper at "X".

ACTUAL CONDITIONS FOUND IN SERVICE

FIG. No. 19.

Not only was the brake shoe thrown out of position but the braking force was largely increased, and being borne entirely by the flange of the wheel, heat checks were developed and large number of wheels removed and replacement demanded of the manufacturer before the cause was determined. A careful analysis of the situation resulted in the remodeling of the brake beams on a lot of 3000 new cars.

FIG. No. 20.

These shoes, reading from right to left, have a bearing surface on the wheel of 69%, 29%, 42%, 55%, 70%, respectively, and show the area of effective shoe surface, as a result of defective shoe and hanger adjustment.

There are certain reasons advanced showing that a hanger a few degrees out of position may be of assistance in regulating the shoe pressure to suit the increased load on the forward wheel which is occasioned by the tendency of the car body to travel ahead of the truck when brakes are applied, having a slight over turning effect which transfers part of the load from the trailer wheel to the forward wheel. This action, however, is so small and the bad effect of too much angle in the brake hanger is so great that the nearer the hanger is maintained at right angle with a radius drawn from the center of the wheel to the center of the shoe, the better. A hanger that is 30 degrees out of position with respect to the normal will theoretically increase the shoe pressure nearly 20%, but as the shoe wears and the hanger becomes slightly distorted, the angularity increases rapidly and the increased pressure on the shoe increases in a geometric ratio above 30 degrees so that a full toggle effect is produced and the brake beam often rises above the center and becomes out of place above a horizontal line through the brake hanger support. The stress on the hanger and on the brake shoe can only be calculated by the ability of the hanger to resist compression stresses. It is very common in cases of this kind to find broken pedestals, and too often, instead of correcting the real trouble, the various parts which become distorted are made of a heavier pattern when repairs are made. There is no point in car construction where such a small item, which often passes unnoticed, can do so much damage.

IRREGULARITIES CAUSED BY IMPROPER PISTON TRAVEL.

The effect of piston travel has a very marked influence over the braking power of a car, especially for partial service applications.

Using the information given in Table No. 2, Page 5, we have the following air pressures in an 8" brake cylinder for various piston travels.

PISTON TRAVEL	AIR PRESSURE IN BRAKE CYLINDER 10 LB. REDUCTION	15 LB. REDUCTION
2"	63 lbs.	63 lbs.
4"	52	59
6"	33	55
8"	22	40
10"	16	30
12"	11	23

TABLE NO. 17.

It will be noted that for a 10 lb. reduction that where cars are properly adjusted, there will be 22 pounds of air pressure, which, for an 8" cylinder represents a total pressure of 1100 lbs. If, however, the piston travel is only 4", which is of frequent occurrence, especially on cars having a high brake leverage, the pressure is 52 lbs. per sq. inch, or a total of 2500 lbs., which is the pressure produced by a full service application at 8" travel, or more than 100% over what was intended. This illustrates the very great irregularity that may exist in braking powers where the piston travel is not properly adjusted, and naturally aggravates conditions to a much greater extent on cars which already have a braking power very largely in excess of the average required for their gross loads.

EFFECT OF IRREGULARITY IN BRAKING POWER ON WHEEL TREADS.

The previous discussions have shown gross irregularities in the distribution of braking power throughout the train, whereby certain cars are called upon to do from 100% to 300% more work in retardation than other cars of the same capacity, and inasmuch as this excessive retardation is transforming mechanical energy into heat the final result must show itself on the tread of the wheel.

Fig. 21 shows the special character of the blemish which is developed from excessive heat between the brake shoe and wheel or between the wheel and rail, when the wheel is skidding.

In the case of wheel A, Fig. 21, the defect was caused by excessive brake friction extending entirely around the circumference of the wheel. The defect in wheel B is largely from the same cause, although not as fully developed. In the case of wheel A, the brake shoe had a bearing near the rim, whereas in B, the brake shoe had a tendency to run as closely to the throat of the wheel as possible. Wheel C represents a series of skidded spots, producing short flat spots and showing that the metal was so intensely heated that thermal cracks appeared and through subsequent pounding the metal has disintegrated.

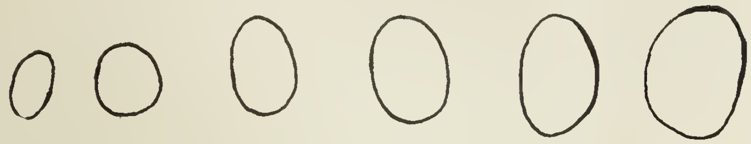
A B C

FIG. No. 21.

Conditions of these kinds occur in bringing a train to a stop, because the greatest frictional resistance between wheel and brake shoe occurs just before the wheel ceases to revolve, and often at this point frictional resistance exists between wheel and rail, in which case the wheel begins to slide. After the wheel once begins to slide, frictional resistance is very much lessened and sliding will be continued until the brake pressure is reduced. In sliding over a distance of only a few feet before the cars come to rest, the term "skidding" is applied, and

a flat or skidded spot, the size of the area of contact between wheel and rail is produced. The following diagram shows actual size and shape of these contact areas under various loads as determined by the American Society of Civil Engineers in a test with new 33" chilled iron wheels on a 75 lb. rail having a top radius of 14".

WHEEL LOAD--Lbs.



5000 10000 15000 20000 25000 30000

FIG. No. 22.

A flat spot no larger than these contact areas is not sufficient to cause the removal of the wheel, but the subsequent pressure and blows received in regular service very often results in the metal shelling out around the center of this contact area, forming a shelled spot as shown in Fig. 23.

TYPICAL SHELL OUT.

FIG. No. 23.

During the time the wheel is sliding all mechanical energy represented in the resistance to motion is transferred to heat through the agency of friction, and as the relation between mechanical energy and heat has previously been explained, the exact amount of heat produced by skidding can be calculated.

For fully loaded cars the heat developed in skidding 10 feet is sufficient to melt the following amounts of iron per wheel:

30 ton car	.59 lbs.
40 ton car	.82 lbs.
50 ton car	1.01 lbs.
70 ton car	1.31 lbs.

This heat is developed in the course of a very few seconds and the amount of metal through which it must pass to the wheel is the mere fraction of an ounce, and being concentrated on one spot the melting point is reached almost instantaneously. The small contact area is subject to a violent expansion which can occur in only one direction, and that is outward from the wheel.

Fig. 24 shows a cross section of a shelled spot. That portion between A and C during the process of skidding is of extremely high temperature, practically at the melting point, and expansion is restricted by the shoulders A and C of cold metal, and as the hottest portion of the metal is at the center B, it expands upward, allowing the concentric rings of metal to expand toward the center, causing a cleavage plane along the line B-D. After the skidding ceases, the metal is very quickly cooled, which causes thermal cracks to be introduced in concentric rings and the subsequent pounding on the rail disintegrates the metal which, after falling out, produces the condition shown in Fig. 23.

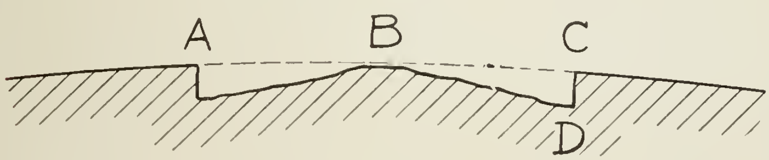


FIG. No. 24.

Where the wheel slides for a greater distance, the melting point is reached and a segment of the metal is rapidly rubbed away which quickly increases the area of contact, giving a much larger surface to receive the heat, thereby reducing the temperature. In this case the heat is sufficient to cause disintegration of the metal from a net work of fine thermal cracks which, as in the case of the shell out, disintegrates and drops from the surface of the tread of the wheel in subsequent service, leaving a rough jagged appearance as shown in Fig. 21-C.

This defect is commonly termed "comby from slid burned." If the sliding had continued over a much longer distance a typical flat spot would have resulted, which would call for the removal of the wheel as soon as it reached a repair track. It was formerly the custom to call these defects "sand holes" or "slag in the metal," indicating an initial defect in the wheel. This idea, however, is not consistent with observations in practice, where it is usual to find such defects in both wheels of a pair and in the same plane, indicating that the defect arose

from slippage on the rail. The tendency of wheels to shell in pairs or in several pairs under the same car is well illustrated in an analysis of results obtained from 500 refrigerator cars, representing a total of 4000 wheels. Of this number, 189 were removed for shelling out, and the relation of this defect to the mate wheel is shown in the following summary:

174 shelled in pairs.
15 shelled singly.
Under 8 cars every wheel was shelled.
Under 4 cars 4 wheels were shelled.

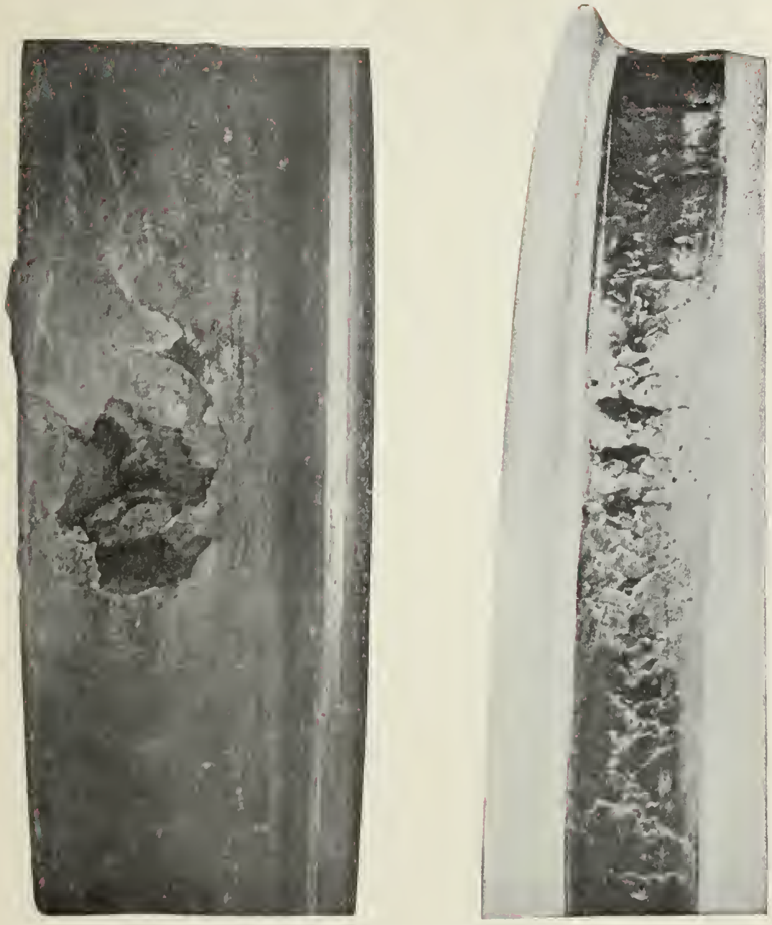
This indicates not only that shelling is due to intense local heating while skidding on the rail, but it is also a matter of observation that this defect occurs under equipment having the highest braking power and making most frequent stops, such as engine tender, heavy passenger and interurban cars; and in freight service it is much more common in the cars of heavy tare weight which, in some cases, have a braking power 10% above the M. C. B. standard. Fig. 25 shows an interesting development of a shell out in which on one wheel a typical shell out has developed while on the mate wheel is a skidded spot. Figs 26 and 27 show thermal cracks in steel wheels.

In the last report of the National Association of Railway Commissioners, in discussing the relation of heat generated by the brake shoe to the tread of the wheel, the following statement is made:

> "So severe are thermal effects of this kind that no grade of steel will resist them, hence the appearance of thermal cracks is not to be taken as an indication that the steel has been lacking in good qualities. Under the influence of heat all metals succomb."

Shows the different development of the shell-out on account of flat sliding. Note the raised center on the left-hand wheel.

FIG. NO. 25.



STEEL-TIRED WHEEL
Shelled.

ROLLED STEEL WHEEL
Scaly Tread.

FIG. No. 26.

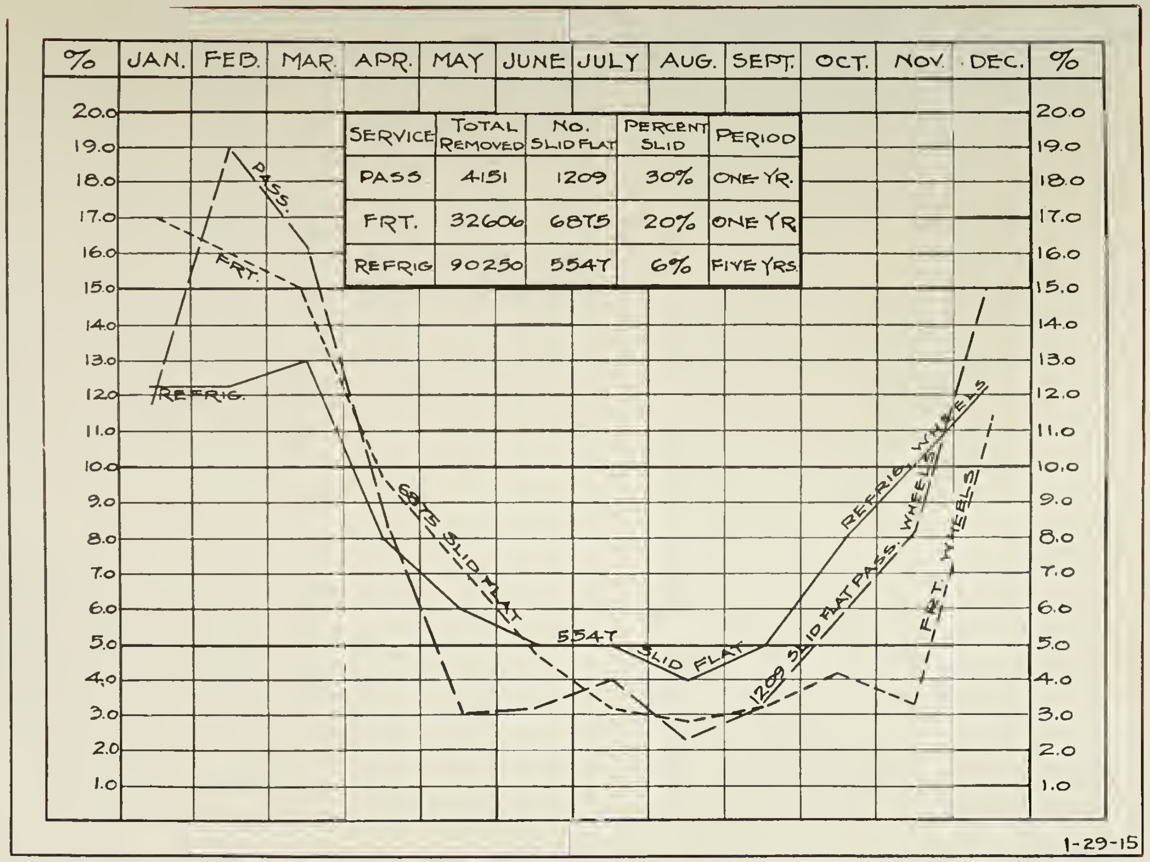
Rolled Steel Wheel—Brake-Burnt, Comby or Scaly.

Fig. No. 27.

Defects of this kind in the chilled iron wheel are not considered elements of danger, for the reason that the thermal cracks and disintegration are largely confined to the chilled metal and do not enter the zone of mottled or grey iron. We know of no case where a wheel has failed on account of a thermal crack.

The relation of braking power to slid flat wheels is very pronounced in cases where the original braking power of the car is calculated at 10% or 15% above the M. C. B. standard and especially where this condition is complicated with a heavy angularity in the brake hanger. Cases are on record where cars of this class have developed slid flat wheels with double the frequency of those in other cars in the same class of service. This indicates that the M. C. B. standard of 60% braking power at 50 lbs. cylinder pressure is about as high as good practice will permit, and where it is desirable to obtain a higher braking power, it is much more feasible to raise the air pressure in the train line which will actuate all cars in the train uniformly, whereas an increase in braking power of a few cars is practically nil, unless such cars can be operated in trains by themselves.

The coefficient of friction of the brake shoe is much more nearly constant than that of the wheel and rail. The condition of rail, whether dry or wet, covered with sleet, dirty, greasy or sanded, creates a wide variation in the coefficient of friction which has a decided effect upon slid flat wheels. This is decisively shown in a record of 1209 slid flat wheels occurring in passenger service and 6875 occurring in freight service and 5547 in refrigerator service, which were tabulated according to the year in which removals were made. The result is shown graphically in Fig. 28, indicating that there are four times as many slid flat wheels in the winter months as in the summer months. This condition, of course, would be aggravated if the percentage of braking power of a car is increased.



SERVICE	TOTAL REMOVED	NO. SLID FLAT	PERCENT SLID	PERIOD
PASS	4151	1209	30%	ONE YR.
FRT.	32606	6875	20%	ONE YR
REFRIG	90250	5547	6%	FIVE YRS.

Fig. No. 28. RELATION BETWEEN WEATHER CONDITIONS AND SLIDING WHEELS

THE EFFECT OF IRREGULARITY IN BRAKING POWER ON TEMPERATURE STRESSES IN THE BODY OF THE WHEEL.

Having shown the effect of excessive and unnecessary heat on the tread of the wheel, the next step is to note the temperature stresses in the plate of the wheel caused by the rapid heating of the tread while the body of the wheel is still cool.

The first indication of excessive temperature stresses in the body of the wheel occurs on mountainous grades, when the tread of the wheel is overheated, which in turn causes a crack in the front plate of the wheel which relieves the tension but calls for the removal of the wheel as soon as the car reaches a repair track. Structural failures of this kind are caused by a temperature stress which originates when the tread is heated suddenly, and before the heat has time to be transferred to the plates. This produces a tensile stress proportional to the difference in temperature. This condition can only arise when the brakes are applied continuously on long descending grades, and does not apply to the application of brakes for stopping the train, for in this case, regardless of the intensity of brake pressure, the time is far too short to produce any serious stress within the wheel.

The heating of the tread is more severe toward the rim than toward the flange, which causes a warpage of the tread, which has a tendency to still further increase the tension in the front plate and creates a compressive stress in the back plate. These are in addition to the stresses arising from flange pressure which reacts on the front plate of the wheel, and centrifugal force which produces a tensile stress in the plates of the wheel. A combination of these stresses is sometimes sufficient to cause rupture in the front plate of the wheel when no attention has been given to the relation between the total stress and the amount of metal required to safely carry the combination of stresses.

The gondola car is always the first to appear in increased capacities. For this reason, wheels for the heavier capacities have always been adjusted to the gondola car, which has a comparatively low braking power. Later on box cars, refrigerator cars, automobile cars of the increased capacities are put into service, when it is found that the heat developed by these cars when placed in mixed trains is sufficient to cause structural failures in the wheels which do not show this defect in the earlier cars of the same capacity.

This is particularly true in the 60000 lb. class, where the range of braking powers within the class is as great as the variation in braking power for all classes.

Observations on various grades and under varying operating conditions very quickly establish the relation of load and grade, which represents the limitation for each class of wheel.

The term "overheating" as now used, is very ambiguous and does not indicate whether the car was designed with excessive braking power; whether the wheel was designed to safely carry the temperature stresses or whether the heat was produced by equipment out of order.

The following facts must be squarely faced when dealing with this subject:

First: The cracking of the plate of the wheel is caused by an expansion stress.

Second: Expansion stresses of sufficient magnitude to crack the plate of the wheel are not produced to any extent except on heavy descending grades.

Third: On heavy descending grades the car having the greatest braking power in a mixed train will produce the greatest amount of heat in the tread of the wheel, and therefore, produces the greatest expansion stress.

Fourth: The temperature stress within a wheel for cars made up of the same class is regulated by the average gross load per car and the length and rate of grade.

The conditions existing at the time the M. C. B. wheels were designed were about as follows:

Maximum Capacity of Car	Capacity of Axle	Gross Load on Four Axles	Tare Weight	Brake Pressure at 60%	Grade Equivalent
60000 lbs.	22000 lbs.	88000 lbs.	28000	16800	3.8
88000	31000	124000	36000	21600	3.6
110000	38000	152000	42000	25200	3.4

TABLE NO. 18

In the above table the maximum capacity of the 80,000 lbs. and 100,000 lbs. capacity cars is given at 10% overload, whereas no overload is calculated for the 60,000 lbs. capacity car, because the overload was not contemplated at the time the 60,000 lbs. capacity axle was designed.

The majority of cracked plate wheels develop under cars in which the above ratios are not followed.

Figs. 29, 30 and 31 show the extreme development in weight of the 60000 lb. class, in which the tare weight runs as high as 55000 lbs. and having a braking power according to M. C. B. standards of 33000 lbs. Reference to Table 15 will show that a braking power as high as 35000 lbs. is encountered in cars having a gross load of 95000 lbs.

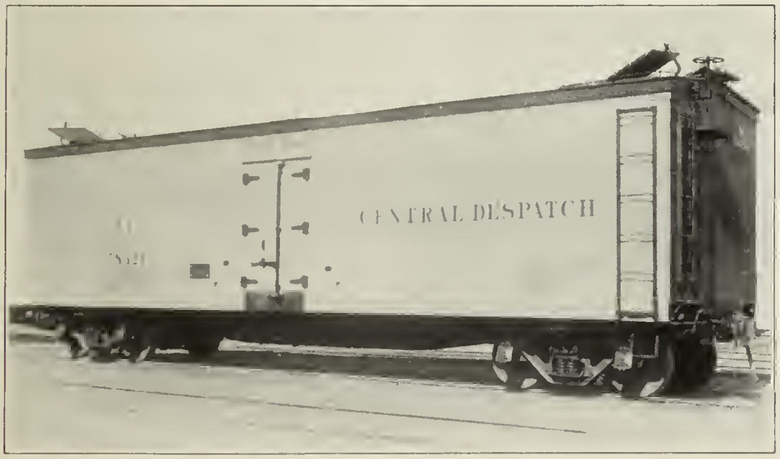


FIG. No. 29.

Capacity 60000 lbs. Tare weight 55000 lbs.

M. C. B. Standard Braking Power 33000 lbs.

Grade equivalent—7.3%.

Heat developed per wheel per minute in terms of pounds of melted iron, at a speed of 30 M. P. H.—10.9 lbs.

Fig. No. 30.

Capacity 60000 lbs. Tare weight 54200 lbs.
M. C. B. Standard Braking Power 32520 lbs.
Grade equivalent—7.2%.
Heat developed per wheel per minute in terms of pounds of melted iron, at a speed of 30 M. P. H.—10.7 lbs.

Fig. No. 31.

Capacity 60000 lbs. Tare weight 48200 lbs.
M. C. B. Standard Braking Power 28920 lbs.
Grade equivalent—6.5%
Heat developed per wheel per minute in terms of pounds of melted iron, at a speed of 30 M. P. H.—9.5 lbs.

Figure 32 shows the other extreme in the same class of cars with reference to gross load, and having scarcely more than one-third the braking power.

FIG. No. 32.

Capacity 60000 lbs. Tare weight 20900 lbs.

M. C. B. Standard Braking Power 12540 lbs.

Grade equivalent—3.0%.

Heat developed per wheel per minute in terms of pounds of melted iron at a speed of 30 M. P. H.—4.1 lbs.

It is not difficult to predetermine which class of car will have the greatest difficulty in maintaining draft gear, brake beams, brake shoes, wheels, etc., especially with reference to wheels, when no provision is made for any variation in temperature stresses.

The same condition is beginning to appear in the 80000 lb. class, as shown in Fig. 33, in which the standard braking power is 40% greater than existed at the time the wheel for 80000 lb. cars was designed.

FIG. No. 33.

Capacity 80000 lbs. Tare weight 51,100 lbs.

M. C. B. Standard Braking Power 30,660 lbs.

Grade equivalent—5.0%

Heat developed per wheel per minute in terms of pounds of melted iron at a speed of 30 M. P. H.—10.1 lbs.

In comparing the above braking pressures with those of the average equipment for the United States, a wide discrepancy is noted for the private cars of the 60000 lb. class, where the average braking power is often larger than that for the average 100000 lb. capacity car.

The average tare weight of the cars of the 60000 lb. class as shown in Table No. 14 is 32,317 lbs. A car of this weight is shown in Fig. 34.

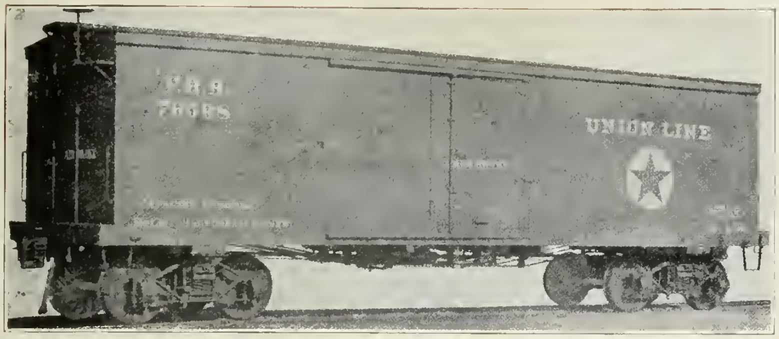


FIG. NO. 34.

Capacity 60000 lbs. Tare weight 32,350 lbs.
This car represents the average weight of all cars of this capacity.

It is necessary in this case to take a box car because the gondola cars in this class are comparatively few, for which reason the average weight in this class is high with reference to the gross load, so that the grade equivalent is 4½%, which is 33% higher proportionately than the average in the 80,000, 100,000 and 140,000 capacity class, respectively.

The average tare weight of the private car lines of the 60,000 lb. class, however, is 40,765 lbs., which is equivalent to a 5.6% grade and is 70% greater in proportion to the gross load than exists in all the cars of heavier capacities.

The average weight of car in the 80,000 lb. class is 36,793 lbs., as shown in Table No. 14. A car of this average weight is shown in Fig. 35.

FIG. NO. 35.

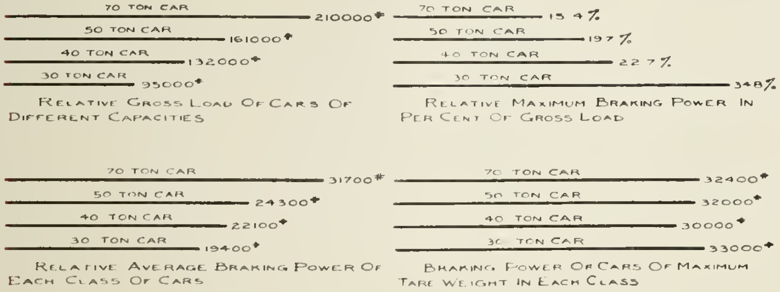
Capacity 80,000 lbs. Tare weight 36,400 lbs.

This car represents the average weight of all cars of this capacity.

The average weight of car in the 100,000 lb. class as shown in Table 14, is 40,429 lbs. and a car of this weight is shown in Fig. 36.

FIG. No. 36.

Capacity 100,000 lbs. Tare weight 41,200 lbs.
This car represents the average weight of all cars of this capacity.



GRAPHICAL REPRESENTATION OF RELATIVE BRAKING POWER OF CARS OF EACH CAPACITY.

FIG. No. 37.

Fig. 37 shows relatively by graphical illustration the gross loads in the four standard classes of M. C. B. equipment; the average braking power of each class of cars; the maximum braking power in each class with reference to gross load; and the maximum actual braking power in each class.

CONCLUSIONS

From the foregoing analysis of the braking power of freight cars, the following questions naturally arise:

Why do cars of heavy tare weight require excessive braking power with reference to their gross loads?

Why is it necessary to go to the added expense of 10" air cylinders with the equivalent brake rigging, brake beams, etc. for equipment in the 60,000 lb. class?

Why should the cars of the 60000 lb. class be required to lend a large percentage of retardation to cars of other classes in the same train?

Why should cars of the private car lines have their maintenance cost increased by excessive brake shoe loss, brake burned wheels and all the other items which grow out of a braking power 70% in excess of that required for cars carrying a heavier tonnage?

Why should the maximum braking power in all classes of cars from 30 to 70 tons be alike?

Why should the heaviest braking power be used on the lightest wheel when it is known that heat stresses are of greater magnitude than any other encountered in service?

If a uniform braking power were used with reference to gross load we would have the following result

Gross Load		Braking Power 15% of Gross Load	Braking Power 60% of Tare Weight	Excess Braking Power in Pounds	Excess Braking Power in Per cent
95000	Private	14250	24500	10250	72
95000		14250	19400	5150	36
132000		19800	22100	2300	12
161000		24150	24300	150	.6
210000		31500	31700	200	.6

TABLE NO. 19.

From the above table it will be noted that the difference in operating conditions for wheels between the 132000 lb. gross load, for cars of 80000 lbs. capacity, and the 161000 gross load for cars of 100000 lbs. capacity is 29000 lbs. in gross load, and practically the same relation between braking power and gross load in each case. To take care of this condition the M. C. B. Association allows 50 lbs. in the weight of the wheel, whereas the difference between the 60000 lb. capacity and 80000 lb. capacity is 37000 lbs. gross load, and an increase of 70% in braking power. In this case also, 50 lbs. variation is allowed in the weights of wheels. This indicates that there is no fundamental rule which covers present standards; also that no condition is fixed in the specification to show the limitation of each weight of wheel, and if the heat stresses are the greatest stresses which the wheel is called upon to withstand and no rule is laid down to show the relation of metal in the plate of the wheel and temperature stress, there is no such a thing at the present time as a standard wheel. We simply have three weights of wheels with no specification as to the service for which they are intended with reference to grade, braking power or any other factor which represents the heating of the wheel, and it is this factor which is more important than the static load factor.

A review of equipment Table No. 14 will show that if the maximum braking power in the 60000 lb. class was based on 37000 lbs., 95% of the cars of this class would not be affected by the rule and that one-half of the cars of the private car lines would also require no change. It would, therefore, seem from every standpoint that the present rule for braking power for freight cars should be amended to the effect that

cars in the 60000 lb. capacity class should have a braking power of 60% of the tare weight at 50 lbs. cylinder pressure for all cars weighing 37000 lbs. or less, and for all cars weighing above 37000 lbs. a uniform braking power of 22,200 lbs. should be used. This would represent an equivalent of a 5.1% grade, and still represent a very materially higher relative braking power than for cars of the heavier capacities.

In the 80000 lb. capacity class, the rule would be that all cars weighing less than 42000 lbs. should have a braking power of 60% of the tare weight at 50 lbs. cylinder pressure, and that all cars above 42000 lbs. should have a uniform braking power of 25,200 lbs., equivalent to 4.2% grade.

All cars in the 100000 lb. capacity class which weigh less than 50000 lbs. should have a braking power of 60% of the tare weight of the car at 50 lbs. cylinder pressure, and all cars weighing over 50000 lbs. should have a uniform braking power of 30000 lbs., equivalent to 4.1% grade.

In the 140000 lbs. capacity class, a braking power of 60% of the light weight of the car at 50 lbs. cylinder pressure for cars under 60000 lbs. tare weight, and for cars weighing over 60000 lbs. a uniform braking power of 36000 lbs. should obtain, equivalent to 3.8% grade.

A rule of this kind would, in a measure, separate cars into classes that would not overlap to the extent which they do at the present time in regard to braking power, and would permit of a condition for designing wheels. A reduction in braking power on 5% of the cars would be unnoticeable as far as retardation of trains goes, and if it was really the desire to maintain the full braking power it would be better to raise the braking power of 95% of the cars 2% than to increase the braking power of 5% of the cars 38%. It would be better still to raise the pressure of air in the pipe line a few pounds, which would operate uniformly on all classes of equipment than to attempt to raise the braking power on a small percentage of equipment. It is understood that if the cars of the private car lines were operated in solid trains there would, of course, be no objection to braking power no matter how high it might be, for the amount of brake pressure that would be used would simply be enough to control the speed on the grade and if all cars in the train were alike there would be no giving and taking of braking power. As the matter now stands, the heavy tare weight cars act as a load brake on the train with no empty brake feature.

If it is believed that there are irregularities in present standards: irregularities in application of the standard; irregularities in brake pressures on long grades; irregularities in brake hangers and irregularities in piston travel, then a concerted action by this Association could accomplish much in correcting these irregularities, the cost of which would be trivial.

The various items should be given publicity and each use his influence within his own sphere of action. These are all items which come under the head of the Safety First program, and a little amount of effort in removing accident hazards would yield a large return to the individual railroad companies.

If there is any one thing that has been absolutely demonstrated it is that the chilled iron wheel can be designed to meet any operating condition now existing or likely to exist on any railroad. It has been conclusively shown that the heaviest stresses that can occur under loads of 25,000 to 30,000 lbs. per wheel can be taken care of with absolute reliability on the heaviest grades in service. Also that under these conditions, structural failures are developed in wheels made of other materials in even greater proportion than they are in the chilled iron wheel.

The Association of Manufacturers of Chilled Car Wheels has for a considerable period of years maintained rules for design of wheels to meet any possible operating condition, and where the conditions are such as to allow freedom in design, perfect results are obtained.

The following information is absolutely essential before any association, committee or any individual can assume the responsibility of recommending standard designs:

First: A summary of the equipment under which wheels are to be used.

Second: Performance of wheels under various loads on various grades.

Third: Relation of stresses in the plate of wheel to various loads on various grades.

Fourth: Relation of the strength of metal to temperature stresses of various intensities.

Fifth: Relation of stresses in the tread of the wheel to load carried.

Sixth: Relation of flange strength to stresses encountered in service.

None of the above items have ever been studied by any association or committee, and for that reason no wheels of the present design can be considered as standardized, and where there is no fundamental information regarding design it is certain that all kinds of errors will creep into specifications. The 675 lb. wheel, for example, calls for 2570 cubic inches of material, weighing .27 lbs. per cubic inch. This amounts to 693 lbs. of material, yet the specifications say the wheel must weigh between 665 lbs. and 675 lbs. This is on a par with designing a receptacle to hold a gallon and sending it to market to be filled, with the specification that but three quarts of material will be paid for. The receptacle is then turned over to the engineer of tests for inspection to see that it is entirely filled. This has been going on for such a long time, and the error is so well known that it is not taken seriously.

In the meantime each individual manufacturer is cutting down the gallon measure according to his own ideas, some removing from the top, others filling in at the bottom, and during the interim with the increasing braking power of cars of the 80000 lb. class, unsatisfactory results are being obtained, all because the specified weight and cubical contents do not correspond. Serious misfits are also in evidence between the tread of the wheel and the frogs in the track. The point of a frog cannot last a week at the proper elevation; the wing rails are cut out in far too short a time because the load is not properly transferred from the point rail to the wing rail. There is a theoretical drop of ¼" every time a wheel trails through a frog, not only causing the frog to wear out in a very short time but the rims of many wheels are chipped to such an extent as to shorten the life of the wheel. It has always been assumed that the flange of the wheel could not be thickened because of interference with the track. This has often been reiterated in the M. C. B. Proceedings. The subject, however, was presented with full data by the Association of Manufacturers of Chilled Car Wheels to the American Railway Engineering Association, and after a very thorough study by a sub-committee appointed to pass on the merits of the question, they came to the conclusion that there is no reason why the flange of the car wheel should not be thickened 3/16". This is a clear indication that it is unsafe for one committee or association to pass definitely on a technical point which is not within their jurisdiction and concerning which they have had no experience, the subject matter belonging to an entirely different department.

In the case of wheel standards, it is necessary to cooperate with the American Railway Engineering Association, the American Society for Testing Materials, the Interstate Commerce Commission, the Bureau of Standards, National Association of Railway Commissioners, the M. C. B. Brake Shoe & Brake Beam Committee.

Any change in standard can be made if there is a good reason for it and the reasons clearly established. However, there seems to be an extreme conservatism in improving standards, such as was displayed in opposing the adoption of the automatic coupler. While this attitude may occasionally prevent the adoption of details that are not entirely satisfactory, it also stands in the way of all improvement and more time and study is given to working up reasons for preventing a useful improvement than in studying the desirable features which might be secured.

The outlook is very promising, as individual railroads are beginning to recognize the fundamentals underlying wheel design and are equipping their individual roads with designs in which service conditions and the metal composing the design are properly correlated.